PRINCIPLES OF INDUSTRIAL PROCESS CONTROL

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FOREWORD

The last thirty years have been a period of great progress in the field of automatic control and automatic regulation. Automatic positioning controllers or servomechanisms are now used in the rapid and accurate control of airplanes, guns, and other fighting equipment.

Automatic control of industrial processes has been of equally great importance in speeding up the production of the many materials necessary in the fighting of a modern war. To mention one example, the synthetic-rubber program would have been impossible without the use of a multitude of automatic temperature, pressure, flow, and liquid-level controllers.

In applying automatic control to industrial processes, however, there are certain fundamental principles which apply to the operation of a process when under automatic control, as well as to the functioning of a servomechanism and its positioned element. This book is primarily devoted to the description and explanation of these fundamental principles.

The author has done an admirable job of drawing together the many loose ends in the literature on automatic control. The complete control system is divided into four elements; the measuring means, the controller mechanism, the final control element, and the process. The pertinate quantities are carefully defined, and appropriate lag coefficients and time constants are used in a treatment of the dynamic characteristics of control systems. Experimental lag coefficients are given for many industrial measuring elements. A number of controlled response curves are given and are used to illustrate the effects to be expected when adjusting a controller on the job. The theory of automatic control is considered with principal emphasis on the controlled and uncontrolled process response curves. The reaction rate-lag method is used to calculate controller settings and reaction rates, and lags are given for a number of representative processes. The cycling period method of setting the reset rate and rate-time controller adjustments is carefully stated and illustrated.

Advancing technology in the industrial field is coming to depend more and more upon precision in processing, introducing standards which would have been thought impossible a few years ago. It is true, there-

fore, that automatic control of industrial processes is in many cases indispensable; in others it is growing in acceptance because of the advances it makes possible in uniformity of product, reduction of production costs, and improvement in quality.

N. B. Nichols

Cambridge, Massachusetts
May 15, 1945

PREFACE

This book is an introduction to the science of automatic control. In order to obtain a working knowledge of the principles of automatic control it has in the past been necessary to refer to a variety of sources. The literature on the subject is widely distributed, and, in addition, it does not completely cover the many phases of industrial process control. Analysis is restricted to more or less highly developed theories applying to particular problems. Since the appearance of Professor W. Trinks' *Governors and the Governing of Prime Movers* in 1919, instrumentation and automatic control have progressed to the development of sophisticated control mechanisms and methods without a parallel development of a generally useful foundation of theory.

The purpose of this book is to treat, in a logical manner, the important laws of operation of industrial automatic control systems and to provide a practical background of theory. Details of measuring devices and controlling mechanisms are brought out only when they are necessary to the analysis of principles and characteristics of operation. The importance of proper measurement is emphasized because of its great influence on automatic control.

Primarily, the book is intended for the student in chemical, metallurgical, mechanical, or electrical engineering. The growing importance of this subject to modern industrial processing has been acknowledged by the addition of courses in instrumentation and automatic control to the curricula of engineering colleges and technical schools. For the student, this early emphasis on automatic control is vital since a process designed and constructed with proper consideration for its control is most likely to be successful. Secondarily, the book may serve as a reference for the industrial user of automatic control equipment.

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Mr. N. B. Nichols of the Radiation Laboratory, Massachusetts Institute of Technology, prepared the Foreword and reviewed the manu-

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DONALD P. ECKMAN

PHILADELPHIA, PA.
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SYMBOLS

C	CAPACITY
c	CONTROL POINT
D	DAMPING FORCE
e	BASE NATURAL LOGARITHMS
F	FLOW
f	FLOATING RATE
G	SPECIFIC GRAVITY
g	ACCELERATION DUE TO GRAVITY
h	HEAD
J	MASS
L	LAG
N	REACTION RATE
P	FINAL ELEMENT POSITION
p	PRESSURE
q	RATE TIME
R	RESISTANCE
r	RESET RATE
S	SPRING FORCE
s	PROPORTIONAL BAND
T	TEMPERATURE
t	TIME
U	TORQUE
V	VOLUME
W	ENERGY
θ	VARIABLE (TEMPERATURE, PRESSURE, FLOW, LEVEL)
π	PI = 3.1416
ϕ	PHASE OR LAG ANGLE
ω	ANGULAR VELOCITY OR PERIOD

CHAPTER 1

THE ART AND SCIENCE OF CONTROL

Automatic control, by virtue of its value to industry, is rapidly assuming importance as one of the newer sciences. It is emerging, however, from the adolescent state of an art dependent upon rule-of-thumb procedures to an exact science based on analytical methods. Fundamental laws and principles are being recognized, and as their significance becomes more widely understood scientific analysis is gradually supplanting the less reliable methods of the past.

An understanding of basic principles is of great value in the analysis of automatic control problems because these principles may be applied to any problem regardless of variations in physical or mechanical details. The control engineer can then recognize the vital factors and readily sift out the unimportant details. The automatic control problem is thereby reduced to its essential components and further analysis is simplified.

Automatic control of industrial processes is but one division of a broad and complex field which includes such diverse subjects as speed governing, temperature control, automatic airplane piloting, automatic machine operation, artillery fire control, and hundreds of other associated subjects. Industrial process control includes the control of temperature, fluid flow, pressure, liquid level, air conditioning, and any other variable quantities of an industrial process.

Electrical circuits and vibrating systems have many elementary characteristics in common with controlled processes. Characteristics similar to those of electrical capacitance, resistance, oscillatory circuits, and other electrical phenomena are found. The forced vibration of a mechanical system and the control of a process variable require almost identical analyses. Industrial process control, therefore, has many analogies in other scientific and engineering fields.

Automatic control is used for the prime purposes of efficiency and economy. It eliminates the element of human error and provides a continuous steady response in counteracting changes in the balance of the process. Automatic control pays for itself in savings of fuel, processing materials, and labor, and in the increased value of the product because of greater output or increased quality.

Automatic control must be properly applied to obtain successful

results. A knowledge of application engineering in automatic control can be secured only by studying the basic laws and principles. This book is intended to serve as a guide to these fundamentals: what they are and how to use them. Many simple devices will be described, but emphasis is placed on theory rather than mechanics.

What is Automatic Control?

Automatic control* can be defined as the maintenance of a balanced state in a process by measuring one of the conditions representing the balance and providing an automatic counteraction to any change in the condition. The balance in the process may be a balance of any form of energy, very often heat or pressure.

For example, the heat balance in a water bath is maintained by measuring the temperature of the water and automatically raising or lowering the flame under the bath. As another example the flow of fluid through a pipe is held constant by measuring the rate of flow and automatically opening or closing a valve so as to oppose any change in flow.

A controlled system is so called because it consists of a process and control system. The process is the operation or function in which a

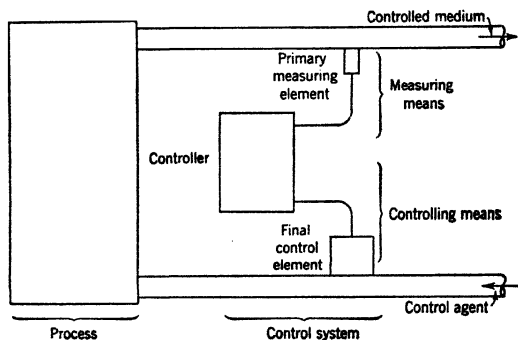



Fig. 1–1. Analysis Diagram for a Controlled System.

balance of conditions is being maintained. Figure 1–1 illustrates schematically the various components of a *controlled system.* Let us take as an example a simple water bath in which a thermometer controller maintains a constant temperature by setting the gas flow to burners under the bath.

Measurement of temperature is accomplished by a thermometer

* A glossary of the terms commonly used in automatic control is included in the Appendix.

bulb which is called a *primary measuring element.* Since the temperature of the outflowing water from the bath is measured, the *controlled medium* is the water in which a *variable* (temperature) is controlled. The primary measuring element is a part of the *measuring means* of the *controller.*

The *controlling means* consists of the controlling mechanism in the controller, such as an electric contact, and a *final control element* which may be an electric motor valve for adjusting the flow of fuel gas to the burner. The fuel gas is a medium for effecting changes in heat flow to the bath and is called a *control agent.* The control agent is always adjusted by means of the final control element.

A *control system* is made up of the measuring means and the controlling means. It generally consists of the controller and all auxiliary mechanisms used for the purpose of controlling the temperature.

The *process* is the operation of heating water to a controlled temperature. The *process equipment* is the physical apparatus, aside from the control system, used in carrying out the process operation. The *process* is the operation or function.

Measurement

The measured variable of the process is generally not an ultimate end in itself but is rather an indication representative of the state of balance of the process. For example, in measuring the temperature of a room the purpose is to measure the degree of comfort of the atmosphere. The temperature is not the only criterion, however; humidity and air motion are also important. The temperature of the room is controlled so that the balance of heat conditions is favorable for personal comfort.

Another example: the pressure in a water system is controlled to insure availability of water in all parts of the system. Pressure is the measured variable, but the controlled condition is the availability of the water supply.

Sometimes the measured variable is purely an indication of a reaction rate within a process. As an example, the outlet temperature of a thermal oil cracking unit is a measure of the rate of cracking. Pressure and other conditions are also important, however. Here, temperature is the measured variable, but the rate of cracking is the controlled condition.

Consequently, as these examples point out, it is necessary to investigate and establish the *meaning of measurement* in automatic control. The question must be continually asked, "Does the measured variable accurately represent the balance of conditions within the process?"

If a fixed, definite relation is maintained between the measured variable and the balance of the process, automatic control will generally be successful. Otherwise, the controller can only maintain the magnitude of measured variable within fixed limits without regard to the final controlled condition.

The purpose of the measuring means is to detect any change or deviation of the controlled variable. This change is then transmitted to the controller for corrective action. Naturally, one cannot expect the controller to counteract for changes which cannot be detected by the measuring means. It follows, then, that a controller is no better than its measuring means.

A change in the measured variable is not instantly detected by the measuring means of any controller. That is, all controllers indicate what the controlled variable *was*, not what it *is*. Thus, we say that the controller has measuring lag. This lag may vary from an extremely small to a very large magnitude and has a most important bearing on automatic control. We shall encounter lag again and again in automatic control, for it is present in the controlling means and process as well as in the measuring means.

Control

Automatic control is accomplished by a circle of events, beginning with a change in the controlled variable and ending ultimately with a forced return of the variable to the desired level. Meanwhile, however, the controlled variable has departed for a period of time from the desired value. Such a change in the state of balance in the process is inevitable because it is the basis for action by the controller.

Let us suppose that the temperature of a water bath is being controlled manually by setting a hand valve to regulate the flame under the bath. If the operator were watching a thermometer placed in the bath, he would open the valve when the temperature goes below the desired point and close the valve when the temperature rises above the desired point. But in order to know when to readjust the valve the operator must observe a change indicated by the thermometer. No matter how closely the operator can read the thermometer, the change in temperature has already occurred before an adjustment of the valve (a corrective action) can be made.

The action of automatic control is dynamic because of the continuous, changing character of the actions within the controlled system. Automatic control rarely achieves a steady state of balance but operates to maintain process conditions within desired limits. It is the

purpose of the controller to hold to a minimum any deviation of the desired balance of the process.

The mode of control is the manner in which the controller responds to a change in the controlled variable. In the example above, the operator could have manipulated the valve in a number of ways in response to a temperature deviation. He might have closed the valve completely when the temperature was above the desired value; he might have closed the valve very slowly at a constant speed; or he might have closed the valve an additional 1 per cent for every degree rise in temperature. Other methods or combinations of methods might also have been used.

The selection of the mode of control is governed to a large extent by the closeness and quality of control desired. It is often poor economy to employ a mode capable of producing a most exacting quality of control if such quality is not required.

The final control element, such as a valve, damper, or electrical relay, is the device which actually guides the control effort — on it depends the coordination of the process and the control system. It is, therefore, a vitally important link in the control system, since the final action of the control system depends on its operation. The operating characteristic of the final element and the mode of control of the controller mechanism together make up the law of action of the control system.

Process

The process comprises an operation or series of operations to which energy is added or taken away in order to maintain a state of balance. For example, in a billet reheating furnace a balance of heat input against loss is maintained by adjusting the flow of fuel so as to compensate for the heat carried away by the reheated metal and for the heat lost by radiation, convection, stack, etc. The state of this balance is determined by measuring a furnace temperature. The process is the operation of reheating the billets.

The operation may not be a process in the chemical sense of the term. The control of fluid flow is an example. Energy in the form of static pressure head is transformed into velocity head when the fluid flows through a pipe. The state of balance is determined by measuring the pressure differential across a resistance to the flow. This simple operation is called a process in automatic-control terminology.

The controlled variable must be truly indicative of the controlled condition in the process. The controlled condition may be defined as

the end purpose for which automatic control is used. The problem is to select the variable which most accurately represents the desired state of balance in the process.

In the example of the billet reheating furnace above, the purpose of applying control is to bring the metal billets into a plastic state so that they may be readily formed by rolling. The plasticity of the billets is proportional to their temperature. Since it is impracticable to measure directly the temperature of the moving billets, the air temperature in the furnace is measured and controlled. The billets must then be left in the furnace long enough to assume the proper temperature.

A process slows down or delays a change in the controlled variable caused by a change in the flow of energy to the process. This slowing down or postponement, called the process lag, is simply the characteristic reaction of dynamic changes in the process.

The lag of a process may be very large or very small. For example, the change in the flow of a liquid caused by a change in valve position in a pipe line occurs almost instantaneously, and the complete change can be made in a matter of fractions of a second. On the other hand, a change made in the temperature of a large annealing furnace by changing the flow of fuel to the furnace may require an hour or more to level out at the new value.

The process, more than any other part of the controlled system, is subject to changes from a number of sources. As many as eight or nine variables may often be found in a process, each one affecting the value of the controlled variable. The desired balance in the process cannot be maintained unless the influence of these auxiliary but related variables is kept at a minimum.

CHAPTER 2

MEASURING MEANS OF INDUSTRIAL CONTROLLERS

Measurement of the variable is the basis for control action since the response of the controller depends upon the detection of changes in the controlled variable. In automatic control there is always a continuous change in the controlled variable. Therefore the dynamic response of the measuring means is equally as important as its static accuracy and dead zone.

Industrial controllers may perform one of two types of service: they may measure and control without indicating the magnitude of the variable; or they may measure, indicate, and control the magnitude of the variable. In some applications there is no need for indication or recording of the variable, and non-indicating controllers are extensively used.

The measuring means of a controller has three functional elements. A primary measuring element, such as a thermometer bulb, thermocouple, or orifice, detects changes in the magnitude of the controlled variable. Transmitting means, such as capillary, wire, or piping, connect the primary measuring element to the controller. A receiving element located in the controller operates the controlling means.

A measuring means may be self-operated, that is, it may use the power developed by its own primary element for operating the controlling means, or it may be power-operated, a source of auxiliary power being utilized to amplify the lesser power of the measuring means. A pressure thermometer, for example, is generally self-operated since it employs the power developed by its receiving element for operating the controlling means. A self-balancing potentiometer is usually power-operated in that the controlling means is operated from an electric motor rather than directly from the receiving element.

It should be emphasized that the performance of any controller and its primary measuring element is largely dependent upon its installation and maintenance. Industrial controllers are precision-made equipment, and their life and performance are greatly improved by sound, well-engineered installation and thorough, methodical maintenance.

To cover in detail all the various phases of the subject of measure-

ment and measuring devices is beyond the scope of a textbook on automatic control. The characteristics important to automatic control will be included, but for a broader study of measurement the reader should consult the references listed at the end of the chapter.

CONTROLLED VARIABLES

Although many conditions may require control in order to maintain the balance of the process, but few actual quantities or variables are measured and controlled.

Temperature is the most important variable to industrial processes. Nearly every process has one or more temperatures associated with its operation. Heat flow may occur in any body, and it is determined by measuring associated temperature differentials. It is not as easily directed, however, as the flow of fluids, because flow of heat does not necessarily involve physical motion of bodies.

Fluid-flow control is a means not only of proportioning the materials introduced into a process but also of supplying energy for purposes of automatic control. For example, the control of feed to a stabilizer column in an oil refinery is necessary in order to maintain the proportion of materials in the column. Another example is found in the control of temperature which may be accomplished by adjusting the flow of steam to the process.

Pressure or *vacuum* control is important in the operation of many continuous chemical processes; it is often associated with temperature conditions. For example, in handling multiple-phase liquids such as water or hydrocarbon compounds, the control of pressure is as important as the control of temperature. Another example is the control of furnace or boiler draft in metal-processing and power plants.

Liquid-level control is essential in continuous distillation processes and in many other industrial operations. Liquid level, however, is closely associated with both fluid flow and pressure. Pressure controllers are also commonly used for the control of liquid level.

Humidity control is important in air conditioning and the processing of foods and textiles. Humidity may be measured by means of dry- and wet-bulb temperatures, partial vapor pressure, evaporation, or physical expansion. Control of humidity by wet- and dry-bulb temperatures is the most common.

Other types of controlled variables in industrial processes are *p*H or acid concentration, gas analysis (CO_2, H_2, O_2, etc.), specific gravity, absolute moisture, and spectrum analysis. The controlled variables most important to industrial automatic control are, however, temperature, flow, pressure, liquid level, and humidity.

PRESSURE THERMOMETER

Thermometers of the pressure-element type utilize the thermal expansion of fluid with increase in temperature to provide an indication of the temperature. If a fluid is confined in a small containing element the expansion of the fluid with temperature raises the pressure inside the element. This pressure, which is proportional to the temperature, is measured. The pressure thermometer is usually a self-operated device.

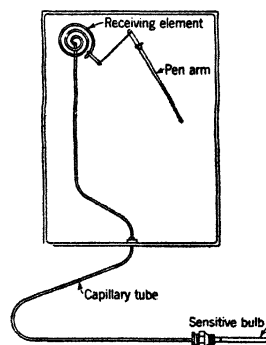
The general construction of a pressure thermometer is shown in Fig. 2–1. A cylindrical bulb filled with a liquid or gas is subjected to the temperature to be measured. A small-diameter tube or capillary connects the bulb with the receiving element in the controller. This receiving element consists of a metallic tube which has been flattened and bent to form a bourdon tube, a helix, or a spiral. If one end of the tube is sealed and the other end fixed, a pressure increase inside the element will cause the sealed end to move in an arc. This motion is utilized to operate the control system.

FIG. 2–1. Typical Pressure Thermometer.

The three types of pressure thermometer controllers are the liquid-expansion, the gas-expansion, and the vapor-actuated. Their difference lies in the various media for filling the thermometer system.

The *liquid-expansion* thermometer is generally filled with mercury, although other fluids such as hydrocarbons are not uncommon. Mercury expansion in a pressure thermometer results in an approximately linear relationship between temperature and movement of the receiving element in accordance with the volumetric expansion equation

$$V_1 = V_0(1 + B\theta) \qquad [2\text{–}1]$$

where V_1 = final volume.
V_0 = initial volume.
B = coefficient of volumetric expansion.
θ = temperature change.

The mercury-filled pressure thermometer is usable over a range of approximately $-35°$ to $1000°$ F.

The *gas-expansion* thermometer uses any relatively inert gas as a filling medium, nitrogen being the most common. The expansion of

a gas follows Charles' law,

$$\frac{p_1}{p_2} = \frac{T_1}{T_2} \qquad [2\text{–}2]$$

where p_1 = initial pressure absolute.
p_2 = final pressure absolute.
T_1 = initial temperature absolute.
T_2 = final temperature absolute.

Since no gas is ideal, the scale of a gas-expansion thermometer is not linear and must be compensated to provide a linear calibration. The range of a gas-filled thermometer is about −200° to 800° F.

The *vapor-actuated* thermometer utilizes vapor pressure to actuate the receiving element. The vapor pressure of a liquid varies with temperature in accordance with the laws of thermodynamics. In general this relationship is not linear because, for equal increments of temperature rise, the vapor pressure increases in progressively greater increments.

The filling media for the vapor-actuated system are many, among them methyl chloride, sulfur dioxide, ether, toluene, butane, propane, and hexane. The range of temperature depends upon the filling medium but is generally about −50° to 600° F.

Liquid- and gas-expansion thermometers require compensation for changes in ambient temperature along the capillary tubing because the fluid in the tubing changes its volume with temperature, thus causing an error. Compensation is achieved in several ways: by means of a bimetallic strip, a volume-compensated capillary, or a second filled system operated either from an additional capillary or from a concentric capillary. These devices correct either the pressure or the movement at the receiving element for changes in ambient temperature in the controller case, along the capillary, or both. The vapor-actuated thermometer does not require compensation.

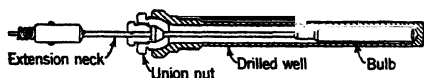



FIG. 2–2. Typical Pressure Thermometer Bulb and Protecting Well.

The speed of response of the liquid-expansion and gas-expansion types of thermometers is about the same under identical conditions, but the vapor-actuated type generally has a slightly higher speed of response. In industrial applications the bulb is usually protected by a socket or well as shown in Fig. 2–2. The fluid whose temperature is being measured surrounds the well, and the heat must be transferred across the well through the air gap and bulb to the filling medium of the

thermometer. The type of well and the method of installation greatly affect the speed of response because the entire mass of metal and fluid in the primary element must be heated or cooled before a change in temperature is indicated.

Special consideration in automatic control must be given to the cross-ambient effect with a vapor-actuated thermometer. When the temperature control point is near the ambient temperature, the speed of response is much slower and the measuring lag may increase by fifty times that at either higher or lower temperatures. This effect is caused by the necessity for vaporizing or liquefying the fluid in the capillary when the measured temperature crosses the ambient or atmospheric temperature. This type of thermometer should not be selected for control of temperatures near ambient unless special provisions are made for improving the response characteristic.

The pressure thermometer is simple in operation, is rugged, and requires very little manual adjustment. It is one of the most widely used of all controllers in the food, chemical, and many other industries.

Thermal-Electrical Primary Elements

A change in an electrical characteristic may be employed as a means of measuring temperature. Primary measuring elements of the industrial type applying this principle are the thermocouple, the resistance element, and the radiation unit. The electrical voltage, current, or resistance is a function of temperature and is measured by a potentiometer or a millivoltmeter controller.

A *thermocouple* is the most widely used thermoelectric element. It is composed of two dissimilar wires welded together at one end. When the temperature at the welded junction changes, an electrical potential is generated in the circuit and appears at the free ends of the wire. This potential is the result of two distinct functions known as the Thomson emf and the Peltier emf.

The Thomson emf is that portion of the total potential caused by the temperature gradient over a single section of homogeneous wire. The Peltier emf is that portion of the total potential produced by the contact of two dissimilar wires. Both potentials vary with temperature.

The thermocouple is used by placing the welded or *hot junction* at the point where the temperature is to be measured. The free ends are carried through extension lead wires to the controller, which point is called the *cold junction*. The controller measures the voltage developed across the open end of the thermocouple. This voltage is caused by increases in temperature of the hot junction over the cold junction. The cold junction thereby becomes a reference point, and the controller

actually indicates the temperature difference between hot and cold junctions.

Several types of thermocouples made up of various combinations of wire are in common industrial use. The table below lists four of these combinations and their characteristics.

Positive	Negative	Recommended Maximum Temperature, °F	Millivolts per Degree F.*
Platinum	Platinum-rhodium	3000	0.0075
Chromel	Alumel	1800	0.0215
Iron	Constantan	1400	0.0290
Copper	Constantan	400	0.0295

The iron-constantan thermocouple may be used at slightly higher temperatures if the atmosphere is not oxidizing. The copper-constantan thermocouple is generally used for low temperatures and may be used below 0° F.

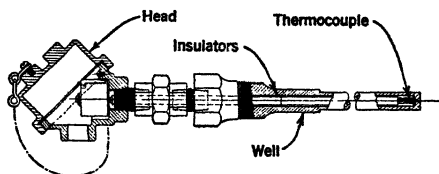



Fig. 2–3. Typical Thermocouple and Protecting Well.

A protecting tube or well between the thermocouple and the fluid whose temperature is being measured, as shown in Fig. 2–3, protects the thermocouple from corrosion and contamination. The well may be made of metal or refractory material, depending upon the installation and the kind of protection required.

The bare thermocouple has an excellent speed of response because the mass being heated is small and the electrical transmission of the change in potential is practically instantaneous. A well materially decreases the speed of response, depending on its size and mass. The iron-constantan thermocouple may have the iron wire in the form of a closed-end tube with the insulated constantan wire running through the center and swaged into the end of the tube to form the hot junction. Such a design reduces mass and speeds the response.

The *resistance element* operates on the principle that the electrical resistance of a wire changes with temperature. A Wheatstone bridge electrical system is the usual device for measuring the resistance, and the controller is calibrated in terms of temperature.

Nickel is the most common material for wire in the industrial resistance bulb. Platinum wire is often employed for greater accuracy or for higher temperatures. The wire is wound on or around a form made of Bakelite, glass, or mica. The coil of wire is assembled in such

* Average value at recommended maximum temperatures.

a manner that it is held firmly in place. The coil assembly is then placed in a sheath or protecting tube and lead wires connect the bulb to the controller.

The accuracy of the resistance element is generally better than that of other forms of temperature-measuring elements. Nickel-wound bulbs are used in the range of 0° to 300° F, although they may sometimes be used below 0° F. The resistance element is quite sensitive to contamination and should not be used in corrosive atmospheres without a protecting well.

The speed of response of a resistance element is usually not as high as a bare thermocouple because of the mass of the bulb and well, and the air spaces within the well. As in the thermocouple, the electrical transmission of the change in resistance is practically instantaneous.

The *radiation unit* employs the principle that the net radiant energy transmitted between two bodies at different temperatures is expressed by the Stefan-Boltzmann law, given by the well-known equation:

$$W = K(\theta_1^4 - \theta_2^4) \qquad [2\text{–}3]$$

where W = net loss in energy per square centimeter.
K = radiation constant.
θ_1 = absolute temperature of radiating body.
θ_2 = absolute temperature of receiving body.

This law is predicated upon the assumption of a black body. A black body is one which absorbs all radiation falling on it and reflects none of the received radiation. Most materials are not perfect black bodies, but this difficulty can be avoided by receiving radiation from the inside surface of a hollow wedge. The ordinary industrial furnace produces the same result as a wedge, and the temperature may be measured satisfactorily by radiation methods. If an error exists by virtue of non-black-body conditions, then the radiation unit is sighted into a target tube with the closed end toward the heated body. The radiation from the open end approximates black-body conditions.

The radiation unit employs a thermocouple or a thermopile (a number of small thermocouples in series) upon which the radiation from the heated body is focused by means of a mirror or lens. Figure 2–4 shows such a unit schematically. An electromotive force is generated when the thermopile is heated by the radiant energy. The scale calibration for the radiation unit is approximately the fourth power of temperature. The radiation unit is usually compensated for the higher temperature inside its housing.

The temperatures between 800° and 3200° are best suited to the

radiation unit. The unit has the great advantage of having no physical contact with the body whose temperature is being measured. However, the presence of hot gases or smoke between the body and the radiation unit may cause an error. Therefore the radiation unit cannot be used where the radiation must pass through absorption media of appreciable concentration.

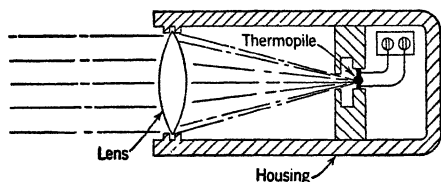



Fig. 2-4. Typical Radiation Unit.

The speed of response of the radiation unit is high because of the very small mass of the thermopile which receives the radiant energy. The electrical transmission of the change in potential is practically instantaneous. The response of the radiation unit is generally slowed when a target tube is used.

Millivoltmeter

As its name implies, the millivoltmeter is a device for the measurement of small voltages. In combination with a thermocouple or a radiation unit it becomes a pyrometer; with a resistance element, it becomes a resistance thermometer. The pyrometer measuring means requires no auxiliary source of power and is therefore self-operated.

Fig. 2-5. Typical Millivoltmeter Pyrometer.

Most industrial millivoltmeter controllers operate on the d'Arsonval principle, that the movement of a coil in a steady magnetic field is proportional to the electrical current through the coil. Figure 2-5 shows the general arrangement of a millivoltmeter. The permanent magnet has two pole pieces fitted around the coil so as to concentrate the magnetic flux in the space around the coil. A coil, suspended on pivots and controlled by hair springs, rotates in the space between the pole pieces. A light pointer attached to the coil indicates the position of the coil. The scale is then calibrated in terms of temperature.

When the millivoltmeter serves as a resistance thermometer, a battery is placed in the circuit and the variation in resistance of the primary element causes a change in the flow of current. A three wire or a five wire connection to the resistance element compensates for lead wire resistance.

The voltage reading of a millivoltmeter is not the same as the voltage developed by the thermocouple because the current flow in the circuit causes a voltage drop over the resistance of the thermocouple, the lead wires, and the millivoltmeter. The controller scale may be corrected by calibrating it arbitrarily at the proper values with certain fixed resistances for each part of the circuit. Any change in resistance of the thermocouple or the lead wires after installation can cause an error. This error is negligible if the internal resistance of the millivoltmeter is high in comparison to its external resistance.

A thermocouple must have its cold junction at substantially constant temperature to provide a starting point for the measurement of change in voltage with temperature. Since the temperature is usually more constant inside the controller than near the point where the thermocouple is installed, the cold junction is extended inside the controller by suitable lead wires. If the ambient temperature at the controller does not change greatly, the cold-junction error will be small. However, a variation in cold-junction temperature may be compensated by various arrangements, such as inserting a bimetallic spiral to readjust the hair spring and correct the error in reading.

The millivoltmeter is simple and inexpensive, and it does not require frequent manual adjustments. It is particularly well suited for the measurement and control of small ovens and heat-treating furnaces. It is also employed as a resistance thermometer for the measurement and control of temperatures in large air-conditioning or refrigerating systems.

Potentiometer

The automatically balanced potentiometer for the measurement of small voltages is operated from a thermocouple or a radiation unit. This instrument is also made in Wheatstone bridge form for use with resistance elements and is then termed a resistance thermometer. Auxiliary power is needed for operating the balancing system, and therefore the potentiometer controller has power-operated measuring means.

The fundamental circuit of the potentiometer is shown in Fig. 2–6. This circuit represents a null type of potentiometer where the unknown voltage is balanced against a known voltage in such a way that zero

current flow at the detecting element or galvanometer indicates balance. The sliding contact can be moved over the slide wire until the unknown voltage of the thermocouple is equal to and opposes a part of the battery voltage. The current flow through the galvanometer or other detecting element *G* is then zero. The position of the slider on a scale can be noted and the controller calibrated in terms of temperature.

The balance is made automatic by utilizing the detecting element to determine the unbalance of the two voltages and to control power to drive the sliding contact, thereby maintaining balance. One example of automatic control is thus met in an automatically balancing potentiometer.

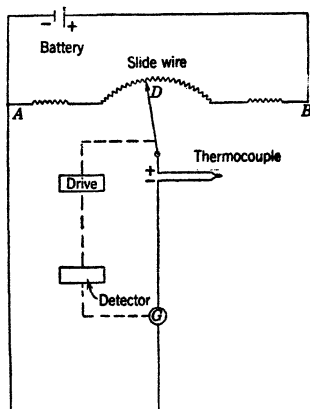



Fig. 2–6. Basic Potentiometer Circuit with Balancing Slide-Wire.

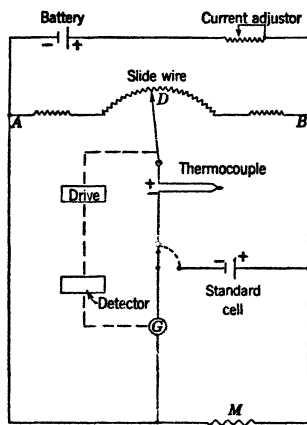



Fig. 2–7. Potentiometer Circuit with Standardization.

The applied voltage must be maintained at a constant value even though there is a constant current drain. Voltage is kept constant in Fig. 2–7 by means of an accurate standard cell, and a current adjustor on the battery voltage. The voltage of the standard cell is compared periodically to a part of the voltage of the battery, and the battery circuit is readjusted to the necessary voltage. This operation, called standardizing, may be done either manually or automatically.

The potentiometer resistance thermometer does not require standardizing equipment since the balance is obtained by matching the resistance of the element at various temperatures with the slide-wire resistance. This balance is independent of small variations in battery voltage.

As with millivoltmeters, cold-junction compensation is also required

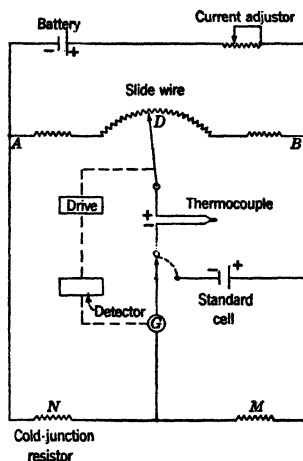
with potentiometers. This is accomplished as shown in Fig. 2-8 by adding a cold-junction compensating resistor N. If the resistor N is made of nickel, which has a temperature coefficient of resistance, any change in ambient temperature will compensate for variations in cold-junction temperature. The other resistors in the circuit may be made of manganin, which has a negligible temperature coefficient of resistance.

Fig. 2-8. Potentiometer Circuit with Standardization and Cold-Junction Compensation.

The mechanisms for detecting and rebalancing the potentiometer, though having many variations, are of three general types. A description of each follows.

The *periodic-balance* potentiometer mechanism has a galvanometer to detect the unbalance of the voltages. When the temperature changes, the electrical circuit is unbalanced and the galvanometer pointer deflects from its zero position, indicating the amount and direction of unbalance. A lever system or step table arrangement is provided to detect the position of the galvanometer pointer while it is clamped momentarily in its deflected position. A lever system operated by a continuously running motor then drives the slide-wire contactor toward the final balance point through a clutch arrangement.

The galvanometer is periodically clamped for a short time at intervals of about 3 seconds. During most of the 3-second period the galvanometer is free to swing to a new position and must be sufficiently damped to arrive at a stable position before it is clamped. The steps taken to arrive at the final balance point are not large enough to permit balancing in one step. Generally the mechanism is arranged to make steps of 70 or 80 per cent of the remaining distance toward the final balance point. In this way the system is made stable.

The *semi-continuous-balance* potentiometer mechanism also has a galvanometer as a means for detecting the unbalance. In one arrangement the deflection is continuously determined by the deflection of a light beam from a mirror on the galvanometer, and a photoelectric cell and amplifier drive an electric motor to rebalance the slide wire. In another arrangement the unbalance is continuously determined by electrical contacts on the galvanometer pointer, and electrical relays

drive the motor to obtain balance. The important advantage of this type is that no direct time delays or waiting periods are introduced in detecting an unbalance.

The *continuous-balance* potentiometer mechanism detects the unbalance by means of an electronic circuit without a galvanometer. The electronic circuit drives the balancing motor and slide-wire contactor directly in rebalancing the potentiometer. The important advantage of this type is that there is no time delay in detecting the unbalance and no lag due to galvanometer action. Balance is obtained without any stepping or periodic action.

The accuracy of any type of potentiometer is independent of variations in the resistance of the thermocouple or lead wires. With the periodic-balance-type potentiometer, however, decrease of external circuit resistance may overdamp the galvanometer sufficiently to cause slower balancing action. This additional lag may be detrimental in automatic control.

The potentiometer is most valuable when precision and accuracy of temperature control are required. It is found in almost every industry.

Flowmeter

The differential or area-type flowmeter is a device for measuring the rate of fluid flow in a closed conduit or pipe. Volumetric meters which measure the volume or quantity rate of flow are not generally employed for control of flow. Practically all flow control is accomplished by a differential flowmeter in which the rate of flow is determined from a measurement of the differential pressure created across a restriction in the line of flow.

The general law for the flow of a fluid is derived from the law of conservation of energy. For turbulent flow, rate of flow is proportional to the square root of pressure differential. It is obvious that flow may be measured by two methods: first, by measuring the differential pressure or head across a restriction of constant area; second, by measuring the area or velocity with a constant differential pressure across a restriction to flow.

The *differential* flowmeter operates on the principle that there is a different differential pressure created across a restriction in the flow line for each rate of flow. The restriction may be any one of the types shown in Fig. 2–9, the orifice plate, the flow nozzle, or the venturi tube.

The orifice plate is the most common form of restriction for flow measurement because it is low in cost, simple to install, and easily changed. Its pressure loss is higher, however, than the flow nozzle or venturi tube.

Pressure taps are located on either side of the orifice plate, and the differential pressure between the upstream and downstream sides is measured by means of a manometer like that shown in Fig. 2–10. The flow nozzle and venturi tube are installed in a similar manner.

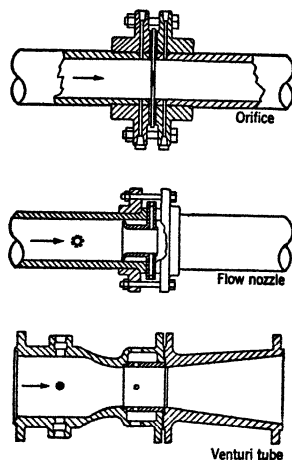



FIG. 2–9. Typical Primary Elements for Flow Measurement.

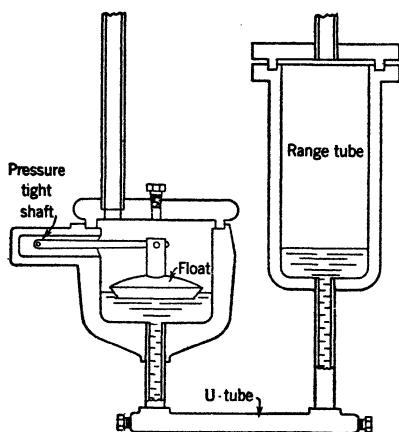



FIG. 2–10. Typical Mechanical Flowmeter Mercury Manometer.

The *manometer* mechanical flowmeter operates from a metal float in an enlarged leg of a mercury U-tube. The metal float follows the changes in level of the mercury in the chamber as the differential pressure changes with the rate of flow. The motion of the float is transmitted through a pressure-tight shaft and is utilized for operating the controlling means.

In the *ring-balance* mechanical flowmeter, the manometer is balanced on a rotating shaft or knife-edge bearings. The differential pressure displaces the mercury in the manometer, and the manometer rotation operates the controlling means.

The *bellows* mechanical flowmeter places the differential pressure across a metallic bellows or diaphragm. The motion of the bellows is transmitted to the controller directly by means of a flexible but sealed connecting rod. In some flowmeters a pneumatic transmitting system connects the transmitter to the controller.

The *bell* mechanical flowmeter operates by placing the differential pressure across a bell suspended in a pool of mercury. The motion of the bell, which is also the float, is transmitted to the controller for operating the controlling means.

Electrical as well as mechanical methods are utilized for the transmission of the motion out of the float chamber to the controller mechanism.

The *inductance-bridge* electrical flowmeter has an armature inside the float chamber positioned by the float and operated in an inductance coil on the outside of the float chamber. An inductance bridge is formed with a receiving coil located inside the controller. The bridge is balanced either by its own electrical forces (self-operated) or by a potentiometric system (power-operated).

The *resistance* electrical flowmeter has a series of resistance wires suspended over the mercury pool in the manometer. When the mercury rises, owing to a change in differential pressure, successive wires ground the electrical current through the mercury. An electrical meter, responsive to the current change, is graduated in terms of flow rate.

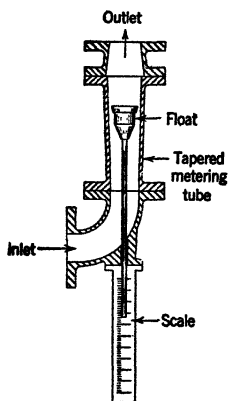



FIG. 2–11. Typical Area Type Flowmeter.

The scale calibration of the differential flowmeter has square-root divisions unless it is especially compensated to produce a linear or semilinear calibration. Linear scale calibration is usually provided by shaping the sides of the mercury manometer or by means of a properly shaped weight on the float.

The speed of response of the orifice-type primary element is practically instantaneous. The flowmeter mechanism, on the other hand, does not have instantaneous response, especially with the mercury manometer, because of the mass and inertia of the parts and fluids in motion. Bellows mechanical flowmeters possess much less mass and their speed of response is greater when measuring the flow of gas.

The *area* flowmeter operates on the principle that there is a different orifice area for each rate of flow and consequently the differential pressure is constant for any rate of flow. This principle is shown schematically in Fig. 2–11. A float is supported in the stream of fluid by the differential pressure across the float. The area for flow is the annular opening between the float and the flow chamber. As the flow increases, the float moves upward, thereby increasing the area of the opening until the differential pressure at the float just balances the weight of the float. Flowmeters operating on this principle usually employ the inductance bridge or other electrical means of transmitting the float position to the controller.

The scale calibration of an area flowmeter is linear if the increase in

area through the meter is linear with various float positions. In the area meter of Fig. 2–11 the area between float and chamber increases as the difference of the square of the float and chamber diameters. The scale is very nearly linear, however, because the calibration is on the flat portion of the parabolic curve of area versus float position.

Pressure Gage

The measurement of pressure or vacuum for purposes of control is generally accomplished by applying the unknown pressure to a movable, elastic member of constant area. The force thus created is balanced by a spring or a weight, and the resulting movement operates the controller mechanism.

The *bourdon tube*, the *spiral*, and the *helix* are the most common actuating elements for the pressure gage as well as for the receiving element in the pressure thermometer. These types of elements do not require a spring since the elastic modulus of the metal is utilized to graduate the deflection due to changes in pressure.

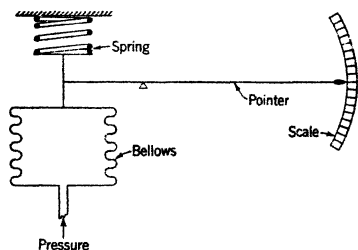



Fig. 2–12. Simple Pressure Gage.

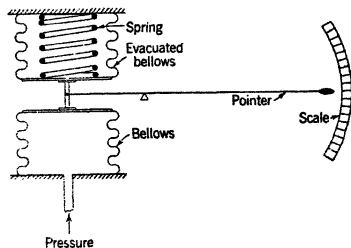



Fig. 2–13. Simple Absolute Pressure Gage.

In the *bellows* pressure gage, illustrated in Fig. 2–12, the unknown pressure is applied to the inside of a bellows. The force of the bellows is balanced against its spring, and the resulting movement indicates the pressure. Absolute pressure is measured by means of a double bellows system as shown in Fig. 2–13. The upper bellows is evacuated to a very low absolute pressure. The pressure to be measured is applied to the lower bellows and is balanced against the force of the evacuated bellows.

The *bell* pressure gage is useful for the measurement of low pressures near atmospheric. The bell is suspended on a beam in a sealing liquid, and the unknown pressure is applied underneath the bell. The force of the pressure on the bell is balanced either by weights or by a spring.

Differential pressures are measured with a second bell at the other end of the same beam.

The *differential* flowmeter is essentially a pressure-measuring device; it is often used without its primary element for the measurement of static and differential pressures.

A pressure gage is quite accurate even though it is self-operated. The primary element in a pressure gage is unique in that it is located inside the controller, whereas nearly all other types of primary elements are located outside the controller.

The pressure gage has a high speed of response because of its comparatively small mass and inertia. The response is materially affected by the size and length of the connecting line and by the viscosity of the fluid in the connecting line.

Liquid-Level Meter

The measurement of liquid level is essentially a measurement of head or pressure.

The *bellows* or *spiral* pressure gage can be connected in various ways. In one arrangement the pressure line from the controller is lowered into the liquid, and the static pressure head creates a pressure in the line and in the primary element. The controller scale is then calibrated for liquid level. The connecting line and element are generally filled with air and sealed by means of a diaphragm so that the fluid cannot restrict or damage the primary element. Another method consists of supplying a small flow of air to the connecting line and allowing it to bubble out into the vessel. The flow of air prevents the fluid from entering the measuring line. Many other arrangements of both the pressure gage and the manometer flowmeter may be contrived for measuring liquid level.

Fig. 2–14. Typical Float-Type Level Unit.

The *float* liquid-level unit shown in Fig. 2–14 generally serves only as a controller with or without indication of the level. The float follows the surface of the liquid, and the movement operates the control mechanism. In another arrangement the buoyancy of the float is utilized to deflect a spring cantilever. This method permits a large change in

level without exceeding the limits of float travel. The interface level between two immiscible liquids of different specific gravity can also be controlled with a float unit. Either of these float-type level controllers requires pneumatic or electrical means for operating the controlling means, and some require it for recording the level.

The *differential-pressure manometer*-type controller, ordinarily employed for measuring flow, is also suitable for measuring liquid level. Since the manometer-type controller measures differential head, it may be used to measure the relative weight of a column of liquid. As long as the specific gravity or density of the liquid is constant, a measurement of liquid level is obtained. If, on the other hand, the liquid level is constant, the specific gravity of the liquid may be measured.

Many electrical types of liquid-level-control devices do not indicate or record the level. An electrode is sometimes introduced to make electrical contact through the liquid and operate a relay for controlling the level. Another type utilizes a column of the liquid to break a light beam and actuate a photocell relay. In some devices a column of liquid serves as a portion of a variable electrical capacitor to operate an electronic control circuit.

Miscellaneous Controllers

A number of other instruments are widely used in self-operated forms or are important as auxiliary equipment.

The *bimetallic thermometer* is constructed by fastening together two metal strips having different coefficients of thermal expansion. As the temperature about the bimetal strip changes, the system deflects to one side or the other. The motion of the strip operates an electrical circuit or sometimes a mechanical controlling means. It is perhaps the most common temperature-indicating device and is simple and inexpensive.

The *expansion thermometer*, responding to the temperature expansion of either mercury or metal, is often employed for the automatic control of temperature. The conventional mercury-in-glass thermometer is made with a contact wire suspended over the mercury. When the mercury column rises, because of an increase in temperature, an electrical circuit is grounded and an electrical relay operates the control circuit. This type will serve as an extremely sensitive controller when the glass bulb is uncovered and the calibration of the thermometer is for a narrow range. The metal-rod thermometer usually operates a controlling means directly as the rod lengthens or shortens as the result of changes in temperature. The speed of response of the metal-

rod thermometer is generally lower than that of most other types because of the high thermal capacity of the rod.

The *expansion hygrometer* utilizes the humidity coefficient of expansion of either a bundle of human hairs or a material such as applewood. The contraction or expansion with changes in relative humidity actuates the controlling means. Neither the accuracy nor the speed of response of this type of hygrometer is very high. For this reason humidity is more often controlled industrially by the separate measurement of both wet-bulb and dry-bulb temperatures by the pressure thermometer or resistance thermometer.

The *moisture-content* meter is valuable for the measurement and control of moisture in cloth or paper. The principle employed is that the electrical resistance through cloth or paper is inversely proportional to its moisture content. The resistance is measured by an electronic amplifier and a self-balancing potentiometer. The controller scale is calibrated in terms of moisture.

Acid-concentration or *p*H meters employ an electrode immersed in the solution so that an electrolytic half cell is formed by virtue of the hydrogen ions in the solution. The potential developed at this electrode is a measure of the hydrogen-ion concentration. Another electrode is added to complete the electrolytic cell, and the potential between the electrodes is measured. One electrode then becomes the reference, and the other the measuring, electrode. The most common reference electrode is the calomel cell, and the most practical of the measuring electrodes is the glass cell. The potential produced by the cells when immersed in the solution is measured with an electronic amplifier and a potentiometer.

The *thermal-conductivity carbon dioxide* meter depends upon the difference in thermal conductivity of the various gases for its operation. Since the thermal conductivity of carbon dioxide is about 40 per cent lower than that of the other components in the flue gas, the thermal conductivity of the mixture is indicative of the carbon dioxide content. A Wheatstone bridge having a constant electric current flowing through the circuit has two of the resistances in cells, one of which is located in a stream of the flue gas and the other in air. The resistance-thermometer measuring device is used with this kind of carbon dioxide meter. An automatic Orsat carbon dioxide meter also serves the same purpose as the thermal-conductivity type.

Analysis meters for hydrogen, oxygen, and explosive gas content generally use the Wheatstone bridge circuit for measurement. Automatic analysis of chemical composition by spectrographic analysis meters is making rapid advances.

Transmission of Measurement

Very often it becomes necessary to locate the primary measuring element at a distance of several hundred feet from the controller. The development of centralized control panels or control rooms has been largely responsible for the increased application of transmission or telemetering methods. Transmission is particularly effective in the measurement of flow and pressure and in many applications may be combined with the pressure-thermometer controller.

Flow controllers installed with lead lines more than 50 feet long always have a slow speed of response owing to the large mass and friction. Pressure-thermometer controllers should never have more than 200 feet of capillary tubing because greater lengths introduce transmission lag and errors of measurement. In such applications, either pneumatic or electric means of transmission improve the performance.

The general features of pneumatic and electrical methods of transmission are:

Pneumatic

1. Explosion proof.
2. Compressed air must be available.
3. Speed of response depends on capacity of pilot and length of transmission line.
4. Versatile in applications with automatic control.

Electric

1. Transmission lines easy to install.
2. Electrical power must be available.
3. Speed of response limited by maximum speed of positioning motor or other device.
4. Speed of response not reduced by longer transmission lines.

Most electrically operated measuring means are satisfactory for transmission by the nature of their construction. The potentiometer controller operating from a thermocouple primary element can have several hundred feet of lead wire between the element and the controller. The inductance bridge and resistance types of flow controllers are suitable for the transmission of flow measurement.

The *Wheatstone bridge* system has a transmitting slide wire located at the primary measuring element and a receiving slide wire in the controller. Either a galvanometer or a sensitive solenoid relay will detect the unbalance of the bridge and drive a balancing motor.

The *Selsyn* or *synchro motor* system consists of two rotatable electric devices similar to synchronous generators. The rotor of one Selsyn is positioned by the transmitting mechanism; the other Selsyn is located in the controller. The movement of the transmitting Selsyn causes unbalanced voltages in the stator circuit, and the receiving Selsyn follows the transmitting Selsyn. The balancing action may be

self-induced, or it may be power-operated when the receiving Selsyn is positioned by an electric motor drive.

The *synchronized rotating-cam* system employs two constantly rotating cams, each driven by a motor, one located at the transmitting point and the other in the controller. A set of contacts is so arranged that the time interval of electrical impulses is proportional to the measured variable. A motor drive is provided to position the receiving element so that the balance of the system is maintained.

The *pneumatic transmission* system has wide application for the transmission of measurement. It consists of an open nozzle supplied with air through a restriction from a source of compressed air, and a flapper positioned by the transmitting mechanism for controlling the back pressure on the nozzle. The back pressure serves to control a large-capacity pilot for producing an air pressure proportional to the measured variable. The transmitter is connected by small-diameter tubing to a pressure gage acting as the receiver. Since this mechanism is almost identical in principle with a pneumatic proportional controller, it is described more completely in a subsequent chapter.

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CHAPTER 3

CHARACTERISTICS OF MEASURING MEANS

Speed of response of the primary measuring element and the measuring means of a controller is the most important single factor affecting the operation of the control system. Since automatic control is a continuous, dynamic function, the rate of detection and the time element in response of the measuring means form an essential part of an analysis. The dead zone, precision, and static error, as well, influence the operation of the control system.

A poor quality of measurement effectively prevents obtaining exact automatic control. If the controller is slow in responding to changes in the controlled variable, or if the detection of small changes is impossible, the controller will be unable to provide the proper counteraction. Thus the failure of a controller is sometimes due to improper measurement.

The method of application and installation of the primary element has as great an influence on the quality of measurement as the design factor. An element designed for fast response may have its advantage completely nullified by improper installation. A primary element very often operates under severe conditions of pressure, temperature, and chemical action. Every precaution must be taken to insure the maintenance of favorable characteristics.

TIME ELEMENT IN MEASUREMENT

Instantaneous response to change in a variable is an ideal that is not likely to be achieved in industrial automatic control. In studying automatic control the first lesson is that the responses of the measuring means, the controlling means, and the process involve the time element. Therefore the *rate* of change is quite as important as the *magnitude* of change.

The time element emphasizes the relative phases by which dynamic changes in the controlled system take place. The time element brings about a phase displacement, particularly in the reaction of a measuring means, and the controller must operate through this time element.

The time element, in a general sense, is called lag. Lag is the falling behind or retardation of one physical condition with respect to some other condition to which it is closely related. For example, a change of temperature at a pressure thermometer bulb is not detected instantaneously. Heat must be transmitted through the bulb wall, and then into

the filling medium, where the change in pressure is transmitted to the receiving spiral. Lag of the pressure thermometer bulb, therefore, involves heat transfer, fluid flow, and pressure transmission in addition to the dynamics of the moving element.

The lag of a measuring means consists of three parts: the lag of the primary element, the lag of transmitting the signal to the controller, and the lag of the receiving mechanism of the controller. No problem in automatic control should be instrumented without due consideration of each of these factors.

Temperature-Measuring Elements

Although temperature is the most important controlled variable in industrial processes, the lag of temperature measurement poses the most problems, owing to the nature of heat transfer and the specific heat capacity of the materials comprising the primary measuring element.

If we were suddenly to immerse the bare bulb of a pressure ther-
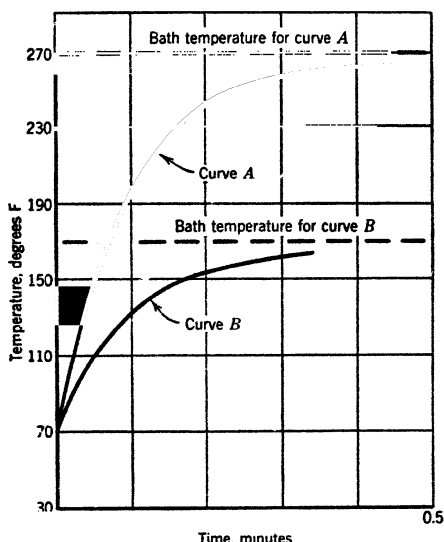
Fig. 3–1. Response Curves for Bare Pressure Thermometer Bulb in Moving Liquid.

mometer in an agitated salt bath maintained at a constant temperature, the thermometer pen would rise as indicated in curve *A* of Fig. 3–1. The curve appears to be logarithmic, and it approaches the bath temperature gradually. At a lower bath temperature the same procedure would yield a curve similar to curve *B* of Fig. 3–1. Thus a thermometer

bulb in an agitated liquid takes as long to respond to a 100° F change as to a 200° F change.

These curves also show that, if it requires 0.25 minute for the thermometer to indicate 90 per cent of the change, it takes another 0.25 minute to reach 99 per cent, and another 0.25 minute to reach 99.9 per cent of the final value. The difficulty of expressing lag in terms of time for indicating a complete change is that theoretically the final temperature will never be reached.

The response curve of a temperature primary element can be determined from Newton's law of cooling, which states[7]* that the rate of change of temperature is proportional to the difference between bath and measured temperatures, or, in equation form,

$$\frac{d\theta}{dt} = \frac{1}{L}(u - \theta) \qquad [3\text{–}1]$$

where θ = indicated temperature.
t = time.
L = lag coefficient.
u = temperature of bath.

If equation 3–1 is rearranged,

$$L\frac{d\theta}{dt} + \theta = u \qquad [3\text{–}2]$$

it may be solved as a linear differential equation of the first order, and

$$\theta = 1 - e^{-\frac{t}{L}} \qquad [3\text{–}3]$$

where θ is the indicated temperature and 1.0 represents the final temperature. Thus for a sudden change in the measured variable an exponential response will be obtained. Although equation 3–3 is arranged so that the temperature starts from zero in order to avoid unnecessary constants, it may be shown that the same relation holds when starting from an initial temperature as in the curve of Fig. 3–1.

It is possible to evaluate the lag coefficient in equation 3–3 because, when

$$t = L$$

then

$$\theta = 0.632$$

* A superior number refers to the corresponding reference listed at the end of the chapter.

Therefore, the time required to reach 63.2 per cent of the total change is equal to the lag coefficient in equation 3–3.

Figure 3–2 shows the response of a mercury thermometer with a bare bulb. The curve reaches 63.2 per cent of the final value in 0.1 minute. The equation for the response of this thermometer is

$$\theta = 1 - e^{\frac{-t}{0.10}} \quad [3\text{–}4]$$

and the lag coefficient is 0.1 minute.
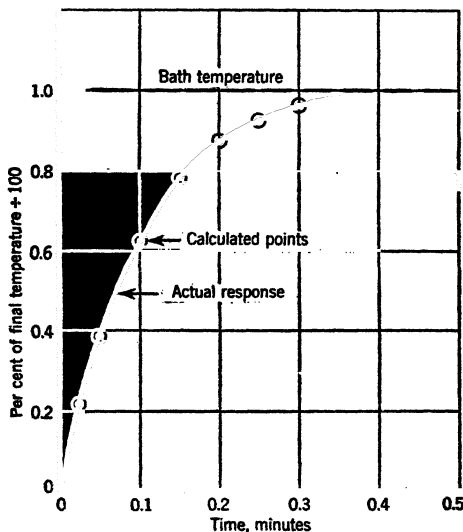
FIG. 3–2. Calculated and Actual Response Curves for a Bare Pressure Thermometer Bulb in Moving Liquid.

If we now substitute various values of t in equation 3–4 the temperature may be plotted against time. The check of the calculated curve and the test curve is shown in Fig. 3–2. It will be seen that the equation approximates the curve quite closely.

The response speeds of the vapor-actuated, liquid- and gas-filled pressure thermometers may be determined by this method. In general, the liquid- and gas-filled thermometers have about the same speed of response and an equal degree of measuring lag. The vapor-actuated thermometer usually has a somewhat higher speed of response and, therefore, slightly less measuring lag.

The lag coefficient for a bare thermocouple can be determined in a simi-

lar manner. Figure 3-3 shows the response curve for a bare chromel-alumel thermocouple when suddenly placed in air in a furnace at constant temperature. From the curve the lag coefficient is found to be 0.58 minute.

Ordinarily, a bare thermocouple has a lower lag coefficient than a bare thermometer bulb. The discrepancy in the two examples is caused by
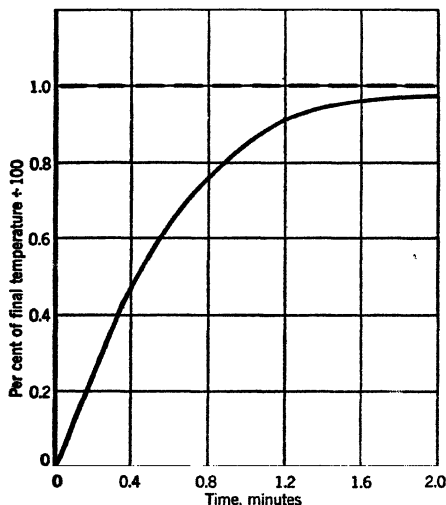
Fig. 3-3. Response Curve for Bare Thermocouple in Moving Air.

testing the thermocouple in air. Air has a much lower coefficient of thermal conductivity as well as a lower thermal capacity than liquid, and the rate of heat transfer is thereby retarded.

The factors influencing the responsiveness of a thermometer bulb or thermocouple are:

Thermal capacity of element.
Thermal conductivity of element.
Surface area per unit mass of element.
Film coefficients of heat transfer.
Mass velocity of fluid surrounding element.
Thermal capacity of fluid surrounding element.
Thermal conductivity of fluid surrounding element.
Time for transmission of change to controller.

The material of the element, such as stainless steel in a thermometer bulb or iron and constantan wire in a thermocouple, determines the rate of transfer to the inside of the element. A larger mass with high thermal

capacity and low thermal conductivity requires a longer time to attain a higher temperature. A larger surface area per unit of mass also increases the rate of heat transfer because it decreases the amount of material to be heated by conduction through the surface.

The movement of fluid surrounding the element is a very large factor in speed of response. If the fluid has very low velocity, a film of cooled fluid will be built up around the element, effectively insulating it. The thermal capacity and conductivity of the fluid determine the amount of heat made available for transfer to the element. Since liquids have higher heat capacity and conductivity than air, a thermometer bulb or a thermocouple in a stirred liquid will respond much faster than one in air.

The transmission lag of a pressure thermometer is dependent upon the length and internal diameter of the capillary tubing. Mercury- or liquid-filled thermometers have negligible transmission lag because of
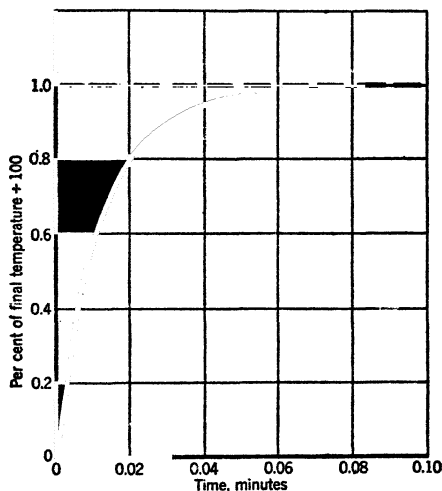
FIG. 3–4. Response Curve for Radiation Unit without Target Tube.

the relatively small volume of liquid required to pass through the capillary on a change in temperature. Gas- and vapor-actuated-type thermometers have a small but appreciable transmission lag because of the compressibility of the filling medium.

Thermoelectric primary elements have practically no transmission lag since very little time is required for electric current to flow through a wire.

The radiation-type primary element receives energy by radiation from the heated source. Figure 3–4 illustrates the response of a radiation

unit. The lag coefficient for this type of primary element is about 0.01 minute. Two factors point directly to this unusual speed of response: first, radiant heat transfer requires practically no time to transmit energy from one point to another; second, the mass of the thermopile in the radiation unit is very small. Response of a radiation unit follows a power law, hence its lag coefficient is only approximate.

The resistance thermometer element generally has a slower response because of the additional mass and slower heat transfer. Figure 3-5
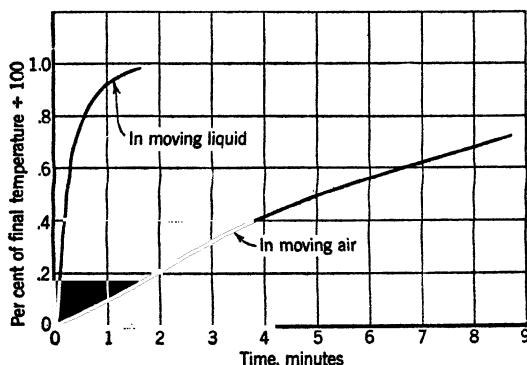
FIG. 3-5. Response Curves for Bare Resistance Thermometer Bulb.

shows response curves for a resistance thermometer element in moving water and in moving air. The important point to notice is the marked difference between the speed of response in the two different fluids.

PROTECTING WELLS AND SPEED OF RESPONSE

In a number of applications it is not possible to expose the bare thermocouple, resistance thermometer bulb, or pressure thermometer bulb to the medium whose temperature is being measured. Any corrosive or oxidizing action is accentuated at higher temperatures and pressures, and it is advisable to protect the thermocouple or bulb against such action by means of a suitable well or socket.

The lag coefficient of a pressure thermometer with a well may be determined by the same method as before. Figure 3-6 illustrates the response curve for the thermometer shown in Fig. 3-2 when fitted with a metal protecting well. The addition of a well has caused the lag coefficient to increase from 0.10 to 1.66 minutes, an increase in lag of about sixteen times the original value.

Likewise, a protecting well may be added to the thermocouple of

Fig. 3-3. The response curve in Fig. 3-7 shows that the lag coefficient has increased from 0.58 to 1.1 for a glass well, to 1.7 for a porcelain well, and to 2.0 for a wrought-iron well. Although iron has a much higher
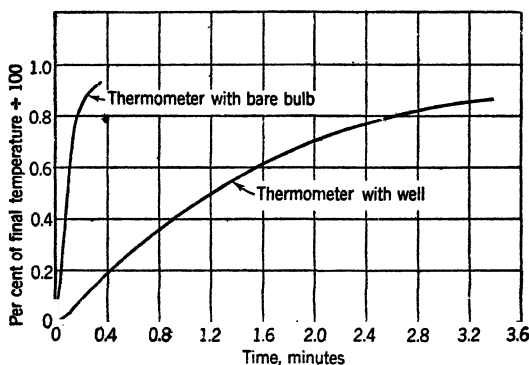
FIG. 3-6. Response Curves for Pressure Thermometer Bulb in Moving Liquid.

thermal conductivity than either glass or procelain, the wrought-iron well has a noticeably higher lag coefficient. On the other hand, the thermal conductivity of glass and porcelain are not as high as that of iron but their smaller heat capacity and more effective use of radiation lower the lag coefficient.
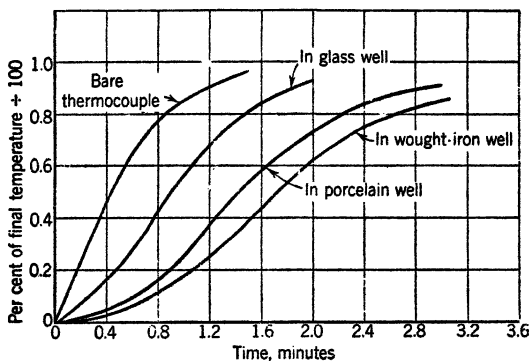
FIG. 3-7. Response Curves for Thermocouple in Moving Air.

The factor of radiation is important in transferring heat at the higher temperatures. For example, one reason why the glass protecting well of Fig. 3-7 has a faster response than the others is that it is translucent and transmits radiation from the heated body directly to the thermo-

couple inside the well. In general, radiation is a much faster method of transmitting heat than conduction. However, if the heated body is at a temperature higher than that of the air surrounding the well, a thermocouple will indicate a temperature somewhere between the two.
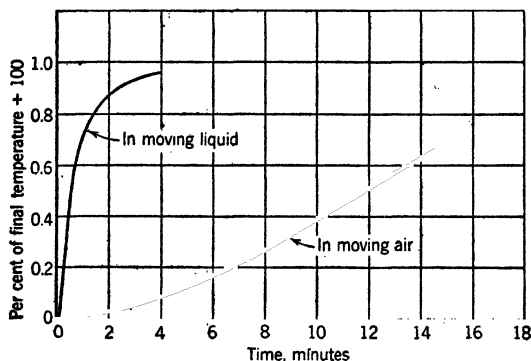
FIG. 3–8. Response Curves for a Resistance Thermometer Bulb with Protecting Well.

A protecting well added to the resistance thermometer bulb previously shown in Fig. 3–5 very greatly increases the measuring lag. Figure 3–8 illustrates the effect of the additional mass and air spaces on the speed of response. The marked difference between the response in air and the response in a liquid should again be noted.
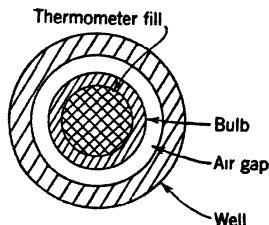
FIG. 3–9. Typical Cross Section of Pressure Thermometer Bulb and Protecting Well.

Air spaces between a thermometer bulb or thermocouple and its protecting well are the cause of additional lag in response of the primary element. A dead air space has insulating qualities and will nullify the advantages gained by selecting materials of low thermal mass and high thermal conductivity. Figure 3–9 shows a transverse section of a thermometer bulb and well. If the bulb does not fit tightly into the well, some means should be provided to obtain metal-to-metal contact. Grounding the tip of a thermocouple to the protecting well to obtain metal-to-metal contact provides similar advantages in obtaining responsiveness.

The response curves of Figs. 3–6, 3–7, and 3–8 for primary elements having protecting wells do not follow exactly the single exponential relation of equation 3–3, as is obvious from the shape of the curve at the

beginning of the change. Whereas the curves for the bare elements start immediately at a maximum rate, the addition of the well causes the response to build up gradually to a maximum rate of change in temperature. The additional lag thus created is of a kind very detrimental to automatic control. The slow response at the beginning of the change does not allow the controller to act rapidly and provide the proper counteraction.

Variation of Temperature-Measuring Lag

The kind of fluid surrounding the element, and the velocity with which it flows, have very great influence on the lag of temperature elements. These factors govern the rate at which a continuous supply of heat is made available for transfer to the measuring element. Low speed of the fluid past the element greatly increases the resistance to heat transfer.

The effect of velocity of the water past a thermometer bulb is shown in Fig. 3–10. The lag coefficient decreases rapidly from 0.10 minute at 2.0 feet per minute to 0.05 minute at 20.0 feet per minute. In order
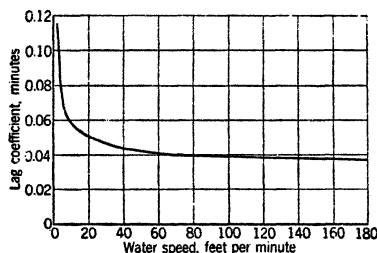
Fig. 3–10. Variation of Measuring Lag with Speed of Water Flow. (From Harper, *Bulletin of the Bureau of Standards*, reference 7.)
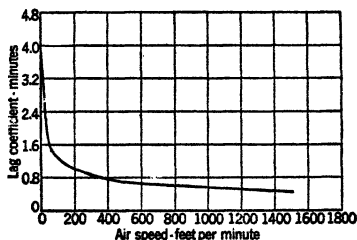
Fig. 3–11. Variation of Measuring Lag with Speed of Air Flow. (From Harper, *Bulletin of the Bureau of Standards*, reference 7.)

to maintain the lag coefficient at a reasonable minimum a liquid should flow past the bulb at a speed of at least 60 feet per minute. If the thermometer bulb or well has rough or uneven surfaces, a speed of 120 feet per minute would be desirable in order to carry away any low-temperature film around the bulb.

The thermal capacity and conductivity of air are low, and a temperature element has a much higher lag coefficient in air than in a moving liquid. Figure 3–11 shows that the lag coefficient for air is greater than that for water. An air speed of at least 400 feet per minute must be maintained in order to reduce the lag coefficient to a minimum.

Radiation plays an important part in the measurement of high temperatures in furnaces. At lower temperatures radiation is not so effective since it follows a fourth-power law. Figure 3–12 shows the
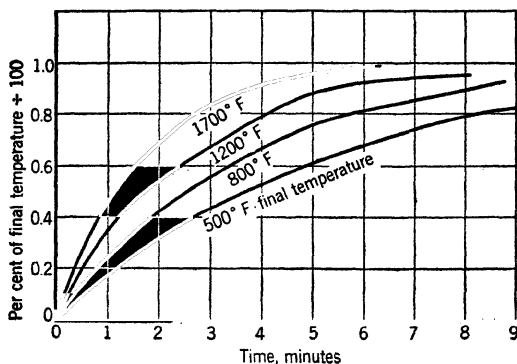
FIG. 3–12. Response Curves for Bare Thermocouple in Still Air with Same Initial, but Different Final, Temperatures.

response curve for a bare thermocouple suddenly inserted into a heating furnace being held at different temperatures. The lag coefficient decreases from 5.7 minutes at 500° F to 1.75 minutes at 1700° F.

A short depth of immersion of a thermocouple or thermometer well,
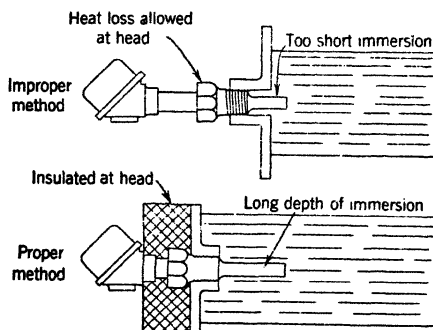
FIG. 3–13. Methods of Installing Thermocouple and Protecting Well.

as illustrated by Fig. 3–13, causes not only inaccuracy but additional lag in response. If the thermocouple and well are not sufficiently immersed in the heated fluid, a low-resistance path for heat flow will be formed along the well toward the outside wall. The heat lost by con-

duction is not available for transfer to the thermocouple. The measured temperature is materially lower than the actual temperature of the air in the furnace. Protecting wells with high thermal conductivity generally must be immersed farther in order to obtain a low lag coefficient.

The radiation-type primary element is not dependent upon actual physical contact between bodies, and for that reason its response is not affected directly by velocity and type of fluid. When a target tube is used, however, sufficient immersion must be obtained so that the end of the tube receives direct radiation. A dirty atmosphere reduces radiation and causes inaccuracy. The target tube for a radiation unit is sometimes immersed in a liquid and the temperature of the liquid is measured by sighting the radiation unit into the target tube. The heat is conducted to the tube and radiated to the radiation unit. In these applications the problems of velocity and type of fluid must be considered.

Dynamic Error and Lag

Thus far in the consideration of measuring lag only an instantaneous change of the variable has been illustrated. Such changes are rarely encountered in actual practice. Consequently, the error and lag of an instrument indicating gradual increases in a variable are important in automatic control.

The response of most measuring elements to a sudden change in actual variable can be approximated by

$$\theta = 1 - e^{-\frac{1}{L}t} \qquad [3\text{–}3]$$

where L varies from 0.01 to 14.0 minutes for the elements here presented. This equation was developed from Newton's law of cooling expressed by equation 3–1, where the bath temperature was constant and the solution was for constant coefficients.

If the bath temperature is a function of time, then a more general solution is required and integration by parts is indicated. The original equation was

$$L\frac{d\theta}{dt} + \theta = u \qquad [3\text{–}1]$$

A general solution is[1, 7]

$$(\theta - u) = (\theta_0 - u_0)e^{-\frac{1}{L}t} - e^{-\frac{1}{L}t}\int_0^t \frac{\partial u}{\partial t}\, e^{\frac{1}{L}t}\, dt \qquad [3\text{–}5]$$

where θ = measured temperature.
u = bath temperature.
θ_0 = initial measured temperature.
u_0 = initial bath temperature.
e = base of natural logarithms.
t = time.
L = lag coefficient.

We may now investigate the lag of indication of a variable when the variable is changing. This more nearly represents actual practice.

Suppose that the temperature of the bath is rising linearly with time, or

$$u = u_0 + kt \qquad [3\text{–}6]$$

then,

$$\frac{\partial u}{\partial t} = k \qquad [3\text{–}7]$$

and the temperature of the bath is rising at a constant rate k.

Substituting in equation 3–5, integrating, and rearranging,

$$(\theta - u) = -kL + (\theta_0 - u_0 + kL)e^{-\frac{1}{L}t} \qquad [3\text{–}8]$$
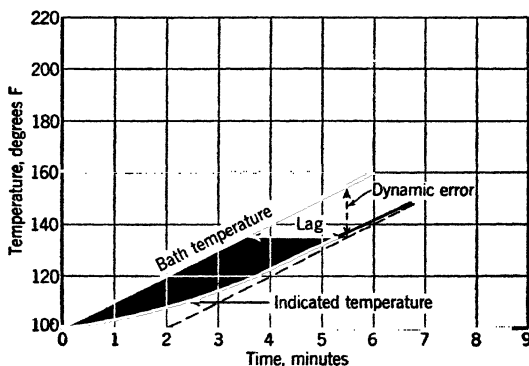
FIG. 3–14. Measuring Lag and Dynamic Error for Element Indicating a Linear Rise of Temperature. (From Beck, Draper, and Bentley, *ASME Trans.*, references 3 and 4.)

Figure 3–14 illustrates both the bath and the indicated temperatures. The curves show that after a short time the indicated temperature lags a constant amount behind the bath temperature. In equation 3–8, if the lag coefficient L is not large, the second term reduces to an insignifi-

cant value. The first term is a constant value, however. Thus,

$$(u - \theta) \rightarrow kL \tag{3-9}$$

Therefore, if the lag coefficient of a measuring means is known, the difference between the true and indicated values of the variable is simply the lag coefficient times the rate of change of the variable. The time difference between identical values of the measured and indicated variables is the lag as shown in Fig. 3–14.

Dynamic error of measurement is always present whenever the controlled variable is changing. In automatic control the dynamic error is usually much more important than any static error of the measuring means because continuous change of the controlled variable about the desired point is often encountered.

Actual examples will show the magnitude of these characteristics. Let us suppose that the temperature of a large furnace is being measured and that the rate of change of temperature is 10° F per hour or 0.167° per minute. If a radiation unit and controller have a measuring lag coefficient of 0.01 minute, the dynamic error would be

$$kL = 0.167 \times 0.01 = 0.002° \text{ F}$$

and the indicated temperature would always be the actual temperature at 0.01 minute previously.

If a thermocouple element and controller have a measuring lag coefficient of 2.00 minutes, the dynamic error would be

$$kL = 0.167 \times 2.00 = 0.33° \text{ F}$$

and the indicated temperature would always be the actual temperature at 2.0 minutes previously. We see from these examples that the dynamic error is not serious when the variable is changing slowly.

Suppose, however, that the temperature of a small furnace is being measured and the rate of change of temperature is 10° F per minute. With the radiation unit, the dynamic error is

$$kL = 10 \times 0.01 = 0.10° \text{ F}$$

With the thermocouple pyrometer, the dynamic error is

$$kL = 10 \times 2.0 = 20° \text{ F}$$

Where the controlled variable changes rapidly, serious errors are introduced when the primary element has a slow response.

Although a controlled variable like temperature does not always change linearly, dynamic error results from delay of the temperature change, no matter what its form. Therefore, a measuring means pro-

duces an indication which is the weighted average of all changes in variable with most emphasis on the lag coefficient period immediately preceding.

Another example may be developed which applies particularly to a type of control that produces a cycling record. Suppose that the bath temperature were cycling with such a relation that

$$u = u_0 + A \cos \omega t \qquad [3\text{–}10]$$

where A = amplitude of bath temperature cycle in degrees Fahrenheit.

$2\pi/\omega$ = period of bath temperature cycle in minutes.

t = time in minutes.

Then

$$\frac{\partial u}{\partial t} = -A\omega \sin \omega t \qquad [3\text{–}11]$$

Substituting in equation 3–5, integrating, and rearranging,

$$(\theta - u) = (\theta_0 - u_0)e^{-\frac{1}{L}t} + A\omega e^{-\frac{1}{L}t} \int_0^t e^{\frac{1}{L}t} \sin \omega t \, dt \qquad [3\text{–}12]$$

If we solve equation 3–12 for the maxima, compare the result to the maxima for the bath temperature, and neglect transient effects, we find that the amplitude of measured or recorded variable is closely approximated by[1]

$$M = \frac{A}{\sqrt{1 + \left(2\pi \frac{L}{b}\right)^2}} \qquad [3\text{–}13]$$

where M = amplitude of cycle of measured variable.

A = amplitude of cycle of actual variable.

b = period of cycle in minutes.

L = lag coefficient of measuring element in minutes.

That is, the amplitude of swing of the measured variable when cycling is smaller than the actual variable.

Figure 3–15 illustrates the cycling of the bath temperature and the resulting indication of the cycle by a controller. The lag in response of the primary measuring element causes the cycle to be delayed as well as reduced in amplitude. The record of the temperature at the controller is indicative of the true temperature or other variable only if the primary measuring element is highly responsive.

Thus, if a temperature is cycling with a total magnitude of cycle of 10° F with a 10-minute period, and the primary element has a lag co-

efficient of 2.0, the indicated or recorded temperature is cycling by

$$M = \frac{10}{\sqrt{1 + 40(\frac{2}{10})^2}} = 6.2° \text{ F}$$
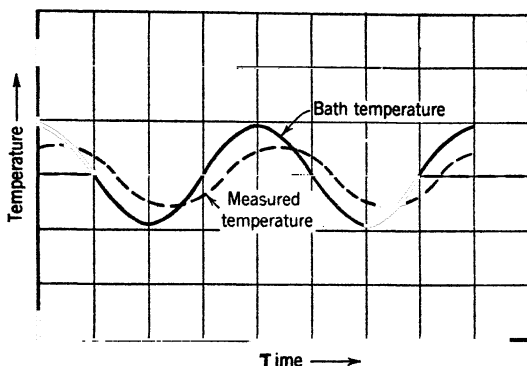
FIG. 3–15. Measuring Lag and Dynamic Error for Element Indicating a Cycling Temperature.

In analyzing the control problem the dynamic error assumes great importance because the record at the controller may not be representative of true conditions. For the best control results it is essential that the measuring lag be as small as possible.

PRESSURE MEASURING MEANS

The characteristics of the action of a pressure measuring means such as a pressure gage, a flowmeter, or a liquid-level meter are different from those of a temperature controller. In most applications pressure changes occur at a much faster rate than temperature changes. Therefore a pressure controller must necessarily have a higher speed of response.

If a pressure is suddenly applied to the measuring spiral of a pressure gage, the resulting movement of the instrument pen is illustrated by the curve of Fig. 3–16. A comparison of this response to the response of the thermometer of Fig. 3–2 shows that the pressure spiral has a very high speed of response. The record of the change in pressure overshoots the final value and oscillates slightly before coming to rest at the higher value.

A response curve for a mercury manometer flowmeter of the type for measuring flow, liquid level, or differential pressure is shown in Fig. 3–17.

The form of the response curve is similar to the response of the spiral pressure gage.

The response illustrated by Figs. 3–16 and 3–17 is in general typical of pressure measuring means. The action is characteristic of a system
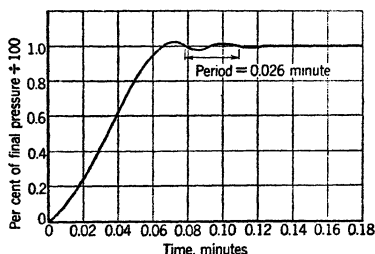
FIG. 3–16. Response Curve for Typical Spiral Element Pressure Gage.
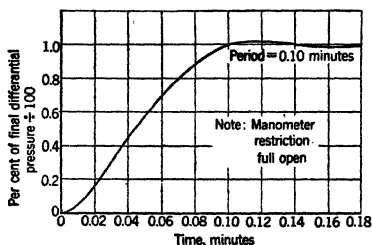
FIG. 3–17. Response Curve for Typical Mercury Manometer Flowmeter.

having mass, inertia, spring force, and damping force. The response of such a system may be described by[4, 10]

$$J \frac{d^2\theta}{dt^2} + D \frac{d\theta}{dt} + S\theta = p \qquad [3\text{–}14]$$

where J = mass.

D = damping force.

S = restoring or spring force.

θ = recorded pressure or motion.

t = time.

p = actual pressure.

The inertia of a pressure gage is represented by the mass or weight of the moving parts of the spiral or other elastic element and pen. The damping force is represented by the fluid friction of the moving parts against the air or the fluid in the spiral. The restoring or spring force is caused by the spring effect of the spiral. Similar forces may be found in the mercury manometer flowmeter.

If equation 3–14 is solved as a linear differential equation with constant coefficients, a particular solution for the oscillatory damped condition is[4]

$$\theta = 1 - e^{-\left(\frac{D}{2J}\right)t} \cos\,(\omega t + \phi) \qquad [3\text{–}15]$$

where θ = pressure (initial value 0, final value 1).
t = time.

$$\phi = \sin^{-1}\sqrt{\frac{D^2}{4JS}} = \text{phase or lag angle.}$$

$$\omega = \sqrt{\frac{S}{J} - \frac{D^2}{4J^2}} = \frac{2\pi}{\text{period}}.$$

The above equation describes the action of a pressure-measuring means when a sudden pressure change is applied. The resulting indication is shown by curve A of Fig. 3–18. Note that the indicated pressure
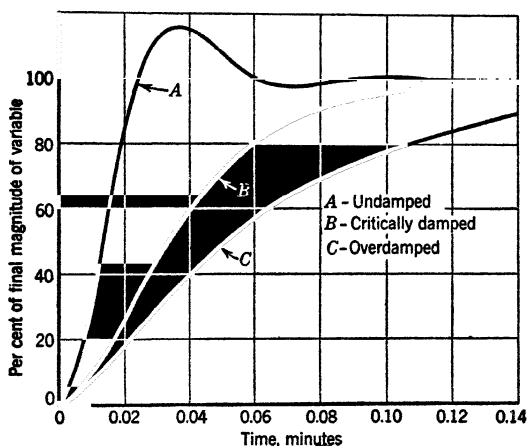
FIG. 3–18. Response of Measuring Devices to a Sudden Change in Measured Variable. (From Draper and Bentley, *ASME Trans.*, reference 4.)

rises slowly at first, then more rapidly until it overshoots the final value and oscillates slightly before coming to rest. This action is, in general, typical of pressure-measuring means when inertia is significant and damping is small.

However, critical damping may be obtained when $D^2 = 4JS$. In this case, the oscillation disappears. Equation 3–15 then becomes, for the same conditions,

$$\theta = 1 - e^{-\frac{D}{2J}t}\left(1 + \frac{D}{2J}t\right) \qquad [3\text{–}16]$$

This condition is represented by curve B of Fig. 3–18, where the indicated pressure approaches the final value in the smallest possible time without overshooting.

When damping becomes too great, overdamping results as shown in curve C of Fig. 3–18. This condition may be described by

$$\theta = 1 - K_1 e^{-K_a t} + K_2 e^{-K_b t} \qquad [3\text{–}17]$$

where
$$K_1 = \frac{1}{2}\left(\frac{1}{\sqrt{1 - (4JS/D^2)}} + 1\right)$$

$$K_2 = \frac{1}{2}\left(\frac{1}{\sqrt{1 - (4JS/D^2)}} - 1\right)$$

$$K_a = \frac{D - \sqrt{D^2 - 4JS}}{2J}$$

$$K_b = \frac{D + \sqrt{D^2 - 4JS}}{2J}$$

A pressure-measuring means may be either underdamped or overdamped, depending on the details of the application. The size and length of pressure connecting lines determine the amount of fluid friction to flow through the lines to the controller. If the lines are long or small in diameter or if the fluid is particularly viscous, the damping force will be increased. This will slow up response and effectively increase measuring lag. Restrictions in the connecting lines should be avoided and the lines should be made large enough to reduce the fluid friction and thereby increase the speed of response.

In the mercury mechanical flowmeter a restriction is usually placed between the two legs of the manometer. Closing this restriction may cause excessive measuring lag due to the added fluid friction and increased damping. Although the orifice itself causes no measuring lag, the piping to the manometer may create excessive friction to fluid flow and thus increase the measuring lag.

The purpose of a manometer restriction as well as a pressure snubber for a pressure gage is to dampen any oscillation or pulsation of the measured variable. Generally these pulsations are rapid and have no bearing on the magnitude of the measured variable. It is important to note that in applying such methods of eliminating pulsation the measuring lag of the controller is increased. For satisfactory automatic control, then, it is better to eliminate the source of the pulsation than to sacrifice responsiveness of the measuring means.

Mass or inertia also has an extensive influence in the dynamic operation of a pressure measuring means. Inertia is caused by liquids in the connecting lines of a pressure measuring means and, in the mercury mechanical flowmeter, by the inertia of mercury in the meter body. Greater inertia causes the system to be underdamped so that greater damping is required in order to obtain a stable measuring means. The increase of both inertia and damping brings about a longer period, and the measuring means may require excessive time for stabilizing. The measuring lag thereby becomes greater.

Long connecting lines to a pressure measuring means of a pressure or flow controller are therefore a disadvantage from two standpoints; both damping and inertia are increased. The resulting increase in measuring lag not only produces greater dynamic error of measurement but also makes automatic control increasingly difficult.

Galvanometer Characteristics

The dynamic action of the galvanometer is an important part of the characteristic of the millivoltmeter and the types of automatically balancing potentiometer which operate from a galvanometer. Generally these controllers are used for temperature where the primary element is likely to have appreciable lag. That the galvanometer lag is small in comparison is shown by Fig. 3–19.

Fig. 3–19. Response Curve for Millivoltmeter Galvanometer.

The dynamic relation for a galvanometer is similar to the action of the pressure-controller measuring means. This characteristic is[7]

$$J\frac{d^2\theta}{dt^2} + D\frac{d\theta}{dt} + S\theta = U \quad [3\text{–}18]$$

where the terms are as before, and U is the applied torque. The solutions to this equation are the same as for a pressure measuring means under the same initial conditions.

The damping of the galvanometer is determined by the electrical characteristics of the coil and magnet, the mass of the coil and pointer, and the torque of the galvanometer spring. Critical damping is nearly always maintained by varying the circuit and shunt resistances of the galvanometer.

The response of a millivoltmeter, and therefore its galvanometer, is shown in Fig. 3–19. The action is generally as near critically damped as possible.

The galvanometer action in a millivoltmeter and an automatically balancing potentiometer is arranged for critical damping with a certain external resistance of lead wire and thermocouple. For example, a galvanometer to be used with an iron-constantan thermocouple 10 feet long and the proper lead wire 200 feet long would have the shunt resistance and spring gradient selected to approximate critical damping at 19 ohms external resistance. If the installation is not arranged so that the external resistance is 19 ohms but is as small as 10 or 12 ohms, the galvanometer will be greatly overdamped and will require a longer time to approach the true reading. A greater external resistance than 19 ohms produces underdamping with attendant oscillatory action.

In studying the millivoltmeter and the automatically balancing potentiometer it was found that the effect of changing external resistance on accuracy is reduced to a minimum. The effect on damping and the resulting lag of indication cannot be compensated. In automatic control, it is important to reduce the measuring lag wherever possible, and the effect of the external circuit resistance must not be overlooked.

Potentiometer Balancing Action

Since the automatically balanced potentiometer is used for exact automatic control more than any other controller, the standards for speed of response are much higher. Generally, every possible precaution is taken to reduce the measuring lag of the primary element. The lag of the controller measuring means then becomes important in comparison.

The galvanometer response in the periodic-balance potentiometer is similar to the pointer action in the millivoltmeter. The characteristic stepping action of the periodic-balance potentiometer is dependent upon galvanometer response and upon the arrangement of the balancing mechanism. The initial deflection of the galvanometer occurs in somewhat less than 3 seconds. The galvanometer is then clamped, its position is determined, and the pen is moved toward the new balance point. The size of the step taken is generally about 70 or 80 per cent of the remaining distance toward balance when there is not a great distance between the initial and the final balance points.

After the first step toward balance is taken, the galvanometer deflection decreases. The entire action is repeated until the final balance point is attained. The number of periods through which the instrument must pass in order to achieve a balance is fairly constant as long as the total change is small.

The dynamic action in balancing is interrupted by periods of rest when the galvanometer is clamped. A direct time delay also occurs at the initial part of the change since the unbalance cannot be detected until a part of the 3-second period elapses.
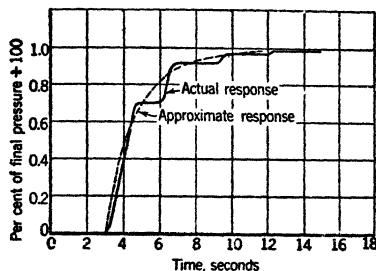
FIG. 3–20. Response Curve for a Typical Periodic-Balance Potentiometer.
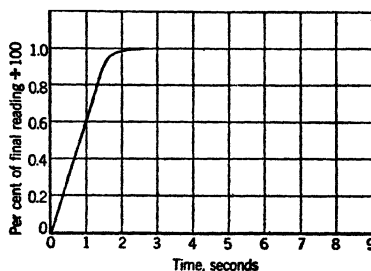
FIG. 3–21. Response Curve for a Typical Continuous-Balance Potentiometer.

The response curve for the periodic-balance potentiometer can be approximated by an exponential curve and a delay as shown in Fig. 3–20. The lag coefficient is about 0.029 minute with an initial delay of 0.03 to 0.05 minute.

The action of the continuous-balance potentiometer in indicating a change is uninterrupted. No initial waiting period is required before the balancing begins. Figure 3–21 illustrates the response of the continuous-balance potentiometer. The semi-continuous-balance potentiometer has approximately this same kind of response except that it may take a short step or two just before it attains the final balance point. The potentiometer is a controlled system, and the balancing action can be only approximated by means of an exponential curve. For purposes of comparison, the lag coefficient is about 0.013 minute.

It is important to note that the time for moving full scale width is approximately the same for nearly all ordinary potentiometer controllers, being in the range of 20 seconds. The response to small changes, such as may be encountered in automatic control, may be greatly different, as shown by the response curves of Figs. 3–20 and 3–21.

Precision and Dead Zone in Measurement

The two factors of precise measurement, static error and reproducibility, must be given their proper weights in automatic control. They are quite dependent upon the application, and each should be considered as an inherent part of the control problem.

The static error of measurement is indicated by the degree of error in determining a specific value of a measured variable. As such it is a static characteristic relating to the manner in which a measurement is made and to the quality of the equipment. Large static error is undesirable but not necessarily detrimental to automatic control.

An example will illustrate this point. In the control of the temperature in a tool-hardening furnace it is impossible to measure any such thing as *the* furnace temperature because the temperature is different in all parts of the furnace, perhaps widely so. The temperature is controlled by selecting one of the points in the furnace as a reference. If the controller has a static error, it is only necessary to determine the proper operating point and then maintain that temperature regardless of the controller reading. The control point of the controller is relied on so that the desired balance in the process may be selected and maintained.

It is sometimes a practice to determine the proper operating points in a pilot plant or small model of the actual industrial plant, and then carry these settings over to the actual application. Sometimes controllers are calibrated in terms of arbitrary values such as 0 to 100. The static error of the controller, when accounted for, shows up as a shift of the operating point, and the process balance is still properly maintained.

Reproducibility of measurement is the degree of closeness with which the same value of a variable may be measured over a period of time. Reproducibility must be obtained if an exact quality of control is desired. In the example above, it might be possible for the calibration of the thermocouple to drift because of contamination resulting from inadequate protection against the furnace gases. This requires periodic shifting of the operating point in order to account for the shift in calibration.

Reproducibility thus becomes more important than static error because it is a time variable and not a static condition. The periodic checking and maintenance of a controller are generally for the purpose of obtaining reproducibility rather than for determining the static error of indication.

The dead zone of the measuring means of a controller is the largest range through which the controlled variable can change without the change being indicated by the controller. In industrial controllers the

extent of this dead zone is usually very small. Dead zone is closely related to sensitivity, and both terms are common.

The principal causes of the dead zone in such self-operated measuring means as the pressure thermometer, flowmeter, or pressure gage are the friction and lost motion of the moving parts of the mechanism. In a well-designed and -maintained controller the dead zone of measurement should be less than 0.2 per cent of scale.

In a power-operated measuring means, such as the automatically balancing potentiometer, the dead zone can be reduced to 0.05 per cent of scale or less, depending upon the method of utilizing auxiliary power to position the indicating element.
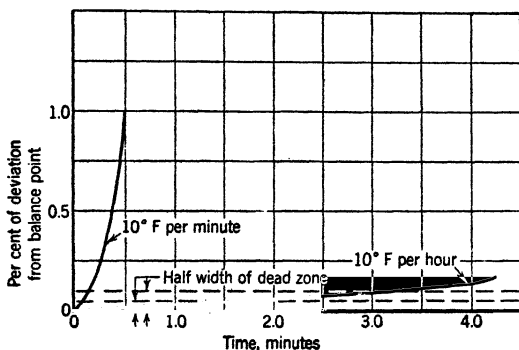
FIG. 3–22. Relation of Measurement Response to Measurement Dead Zone.

The effect of the dead zone of measurement is to create a time delay in providing the initial impulse to the controller. Figure 3–22 illustrates this action. If the variable has previously balanced within the dead zone, the change in variable takes place as shown by the curve. The initial change will not be detected by the controller until the controlled variable has reached the edge of the dead zone. A length of time, depending upon the rate of change of the variable, will have elapsed. This type of delay is most serious because the controller cannot make an early correction for the change.

The length of the time delay, or dead time as it is called, may be determined from Fig. 3–22. If the temperature is changing at a rate of 10° F per minute, then a dead zone of 0.1 per cent causes a dead time of 0.1 minute, and a dead zone of 0.2 per cent causes a dead time of 0.15 minute. If the temperature changes slowly, the same dead zones cause dead times of 1.5 and 3.3 minutes. Therefore, the slower the change of the variable the more serious a measurement dead zone becomes.

Summary of Lag Coefficients

In order to provide a relative comparison between the various controller measuring means and their primary elements, all the lag coefficients previously used are listed in the table below. This table shows the order of measuring lags encountered in actual practice and the wide range of these lags. It should be remembered that any one of these lag coefficients may vary considerably and that these figures are not indicative of either the smallest or the largest lag that can be obtained.

Type of Element or Controller	Bare or in Well	Moving Liquid or Air	Minutes Lag Coefficient
Pressure thermometer	Bare	Liquid	0.10
Pressure thermometer	Well	Liquid	1.66
Resistance thermometer	Bare	Liquid	0.50
Resistance thermometer	Bare	Air	7.00
Resistance thermometer	Well	Liquid	0.70
Resistance thermometer	Well	Air	14.00
Thermocouple	Bare	Air	0.58
Thermocouple	Glass well	Air	1.10
Thermocouple	Porcelain well	Air	1.70
Thermocouple	Iron well	Air	2.00
Radiation unit	Without tube	Air	0.01
Pressure gage	....	....	0.03
Flowmeter (mercury manometer)	....	....	0.04
Millivoltmeter	Without element		0.01
Potentiometer (periodic)	Without element		0.03
Potentiometer (continuous)	Without element		0.01

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Chapter 4

MODES OF AUTOMATIC CONTROL

The controller interprets the indication of changes in the variable supplied by the measuring means and produces the counteraction necessary to maintain the desired balance within the process. The method by which the controller produces a counteraction is called the mode of control. This mode, together with the lag and the dead zone, constitute the controller characteristics.

Each type of controller has a clearly defined mode, generally one of the following:

Two-position
Single-speed floating
Proportional-speed floating
Proportional
Proportional-reset
Proportional-reset-rate

In analyzing a specific control problem, a choice must be made among the various modes of control. Generally speaking, the more difficult the control problem, the farther down the list one must go in order to achieve the desired result. This does not mean that a complicated controller is necessary to obtain good control; on the contrary, the simplest mode is often capable of providing a high quality of control.

Many of the modes of control operate through either pneumatic, electric, electronic, hydraulic, or mechanical means. The pneumatic and electric forms are the most common in industrial process control. It is quite simple to relate the characteristics of the pneumatic and electric forms of controllers to the other forms.

Two-Position and Multiposition Modes

These types of controller action, as the names imply, are ones in which the final control element, such as a valve, is quickly moved to only one of two or three positions when the controlled variable reaches a predetermined magnitude. The simple thermostat is an example of the position mode. When the temperature is above the desired point the valve is closed, and when the temperature is low the valve is opened. The two-position and multiposition modes, therefore, do not recognize

rate or magnitude of deviation but react only to fixed magnitudes of the controlled variable.

The three common types of position controller action are the two-position, the two-position differential, and the three-position. Occasionally there are more than three positions.
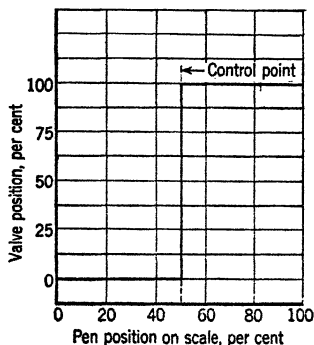
FIG. 4–1. Action of Two-Position (On-Off or Open-and-Shut) Mode of Control.
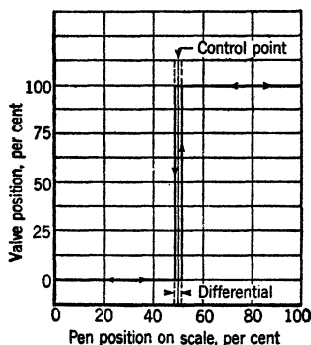
FIG. 4–2. Action of Two-Position Differential Mode of Control.

The *two-position* mode, often called on-off, is a position type of controller action in which the valve is quickly moved to either the closed or open position, depending upon whether the controlled variable is above or below the control point. Figure 4–1 represents this action schematically. Pen motion above and below the control point operates the valve. If the valve or other final control element has a means for limiting its travel, the two-position controller may move the valve between partially open and partially closed positions.

FIG. 4–3. Action of Three-Position Mode of Control.

The *two-position differential* mode is a two-position mode in which the final control element or valve remains in its last position until the controlled variable has moved slightly beyond the control point. This action is shown schematically in Fig. 4–2. When the temperature, for example, is rising, the valve remains open until the pen is slightly above the control point. When the temperature

is falling the valve remains closed until the pen is slightly below the control point. Thus the valve remains in its last position through the differential gap.

The *three-position* mode is a type of controller action in which the valve or other control element is positioned in one of three settings, depending upon whether the controlled variable is above, close to, or below the control point. Figure 4-3 illustrates the three-position mode. When the temperature is in a position near the control point, the valve is set to a partially open position. When the temperature moves out of the middle position the valve is either opened or closed.

Two-Position Controller

The most widely used two-position controllers are of the electric or electronic types. Control is obtained by means of either an electric power relay, a solenoid-operated valve, or a motor-operated valve.
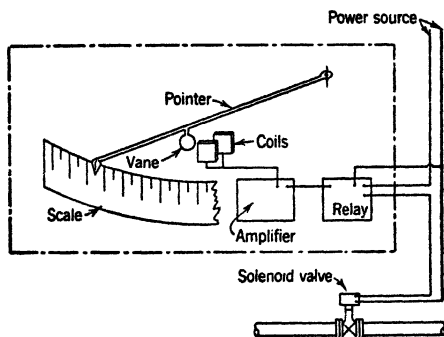
Fig. 4-4. Typical Two-Position (On-Off) Controller.

The controller may be a pressure thermometer, a millivoltmeter, or a potentiometer. The control mechanism is arranged so that electrical contacts are made and broken when the controller pen or pointer passes the control point. These are usually called the high contact and low contact if two are required. Figure 4-4 illustrates schematically one of the many types of electric and electronic controllers.

The differential between the high and low positions constitutes a very small percentage of full scale unless the controller is intentionally constructed with a differential. A differential of zero width, such as knife-edge operation approaches, is very difficult to obtain. In most electric two-position controllers the differential gap is about 0.1 per cent of full scale.

A differential is often used intentionally in the controller in order to lessen wear on the control mechanism resulting from frequent operation. The differential is then set for whatever width is desired but is generally less than 2 per cent of full scale.

Fig. 4-5. Control Characteristic of Two-Position Controller Using an Electric Motor-Operated Valve.

Controller lag is the retardation in the response of the final control element, such as a valve, to a change in the recorded or indicated variable at the controller. If a two-position electric controller is used to operate a power relay or a solenoid valve, then the controller lag is the time which elapses between a making or breaking of the contact in the controller and the subsequent operation of the valve or relay. The controller lag of the electric or electronic controller with a relay or solenoid valve can generally be neglected.

The controller lag of a two-position controller and a motor-operated valve may be large because the motor requires a definite time at its running speed to move from one position to the other. A motor-operated valve usually requires less than 120 and more than 2 seconds for full stroke. Figure 4-5 shows the action of a motor-operated valve on control.

Pneumatic on-off controllers are also widely used. This type will be discussed under pneumatic proportional controllers.

Many self-operated controllers are of the two-position type. Their operation is not essentially different from that of the electric or pneumatic two-position controller.

Single-Speed and Multispeed Floating Control

In the floating mode of control the final control element is gradually moved toward either the open or closed position, depending upon whether the controlled variable is above or below predetermined magnitudes. In other words, the valve is gradually closed or opened and floats in a partly open position. The three types of floating are single-speed floating, single-speed floating with neutral, and multispeed floating.

The *single-speed floating* mode does not recognize rate or magnitude of deviation but reacts only to the elapsed time of the deviation. The speed of floating is constant, regardless of the magnitude of the deviation.

Single-speed floating control may be achieved by means of the same equipment as for two-position control with an electric-motor-operated valve. The difference lies only in the speed of operation of the motor. In two-position control the motor valve timing is 120 seconds or less. In single-speed floating control the timing for valve stroke is usually 120 seconds or more. An electrical interrupter is sometimes employed in conjunction with the motor to decrease the speed of opening and closing the valve.
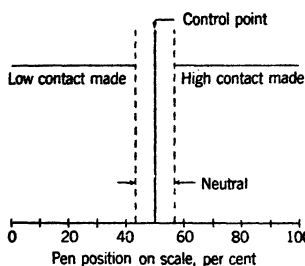
FIG. 4–6. Controller Arrangement for Single-Speed Floating with Neutral Control.
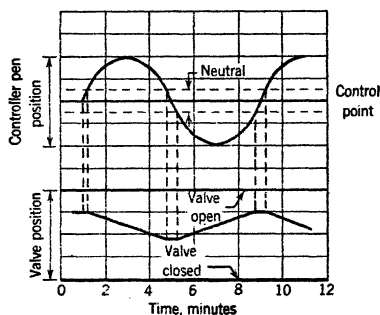
FIG. 4–7. Control Characteristic of Single-Speed Floating Controller with Neutral.

Single-speed floating with neutral controller operates in the same manner except that a neutral is employed in the controller so that motor reversals are not so frequent. The neutral in the controller shown in Fig. 4–6 is arranged so that no contact is made when the pen is near the control point. The motor-operated valve is stationary when the controlled variable is close to the control point. This action is shown by Fig. 4–7. A neutral is customary with single-speed floating control.

Another type of single-speed floating with neutral controller is made for controlling electric current to an electrically heated furnace. Figure 4–8 shows a schematic arrangement where the controller automatically adjusts the percentage of time that the load contact is made during 1-minute periods. The constantly rotating motor determines the basic period. The reversible motor moves only when the temperature is above or below the neutral. The resulting action of the power relay is to supply electric power to the furnace at a variable rate.

Controller lag in a single-speed floating controller is insignificant since valve movement continues in one direction or the other until the temperature again approaches the control point. The dead zone
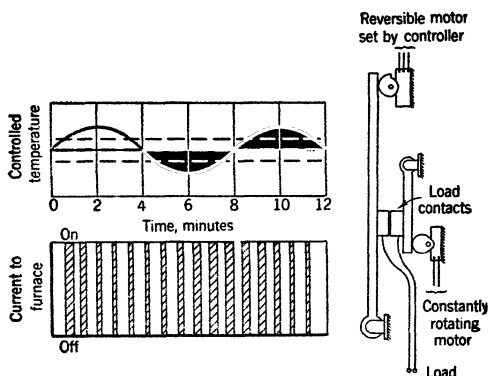
FIG. 4–8. Schematic Diagram and Control Characteristic of Single-Speed Floating Controller for Electric Heating.

of the electrical controller for this mode is usually about 0.1 per cent of full scale unless a neutral is used.

The *multispeed floating* mode is a type of controller action in which the final control element is given any one of several speeds depending upon whether the controlled variable is close to or far from the control point. Thus, this type of controller recognizes elapsed time of deviation and, to a limited extent, the magnitude of deviation.

FIG. 4–9. Action of Multispeed Floating with Neutral Mode of Control.

The action of a two-speed floating with neutral controller, which is the most common type, is shown in Fig. 4–9. When the controlled variable is in the neutral at the control point the valve is stationary. If the controlled variable moves out of the neutral the valve moves at a slow constant speed, and upon greater deviation moves at a fast constant speed.

The multispeed floating controller is generally an electric type, having controller contacts to drive a reversible valve motor. The fast speed is usually the maximum speed

of the motor; the slow speed is generally obtained by means of an electrical interrupter which supplies power to the motor for fractions of a fixed cycle when motion is required. The interrupter is dropped out of the circuit when the fast speed is required.

Proportional-Speed Floating Control

In this type of controller action the final control element, such as a valve, is moved at a rate dependent upon the magnitude of deviation of the controlled variable from the control point. For example, if the controller pen rises a certain amount above the control point, the valve begins to close at a constant rate. If the deviation becomes twice as great, the valve moves twice as fast. When the variable is at the control point, the valve remains stationary. Thus, a propor-
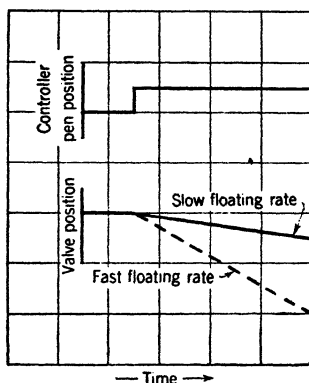
Fig. 4-10. Response of Proportional-Speed Floating Controller to a Sudden Change.
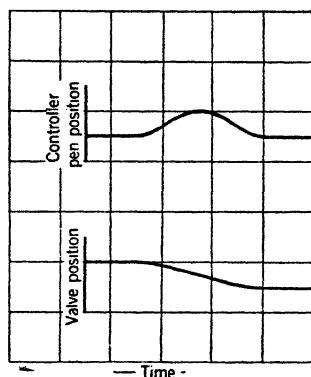
Fig. 4-11. Response of Proportional-Speed Floating Controller to a Stable Change.

tional-speed floating controller recognizes not only elapsed time of deviation but also magnitude of deviation.

Although the proportional-speed floating mode may be used by itself, it is very often combined with the proportional mode, when it is known as reset response. Therefore, an understanding of its operation and its action is important.

In order to study the dynamic action of a controller, we may assume that the controller is not applied to a process and that certain movements of the pen (corresponding to changes in measured variable) result in different changes in controller output signal. By studying the relationship between controller input and output, its mode of control is determined.

The response of the controller to a step change is shown in Fig. 4–10. When the variable rises suddenly to a new value, the valve begins to move at a fixed rate. Most controllers of this type have an adjustment of floating rate. This figure shows that, by adjusting the floating rate, the valve may be made to move at almost any speed for the same deviation of controlled variable from control point.

Proportional-speed floating action is identical with an integrating response since both elapsed time and deviation are recognized. In Fig. 4–10, it will be seen that the magnitude of valve position represents the area under the deviation curve. Since the area increases linearly, the valve position changes at a constant rate.

The controller response to a stable change is shown in Fig. 4–11. The controlled variable deviates gradually and then returns to the control point. The controller responds to area under the deviation curve, and the valve finally settles in a different position. Note that the rate of motion of the valve depends upon the deviation. At the point of greatest deviation, the rate of valve motion is the greatest.

The controller equation for the proportional-speed floating mode may be determined from the responses shown in Figs. 4–10 and 4–11. The rate of valve motion depends upon deviation. Therefore, assuming a linear controller scale

$$-\frac{dP}{dt} = f(\theta - c) \qquad [4\text{–}1]$$

where P = valve position in per cent divided by 100.
f = floating rate in per cent per minute valve motion per per cent deviation.
θ = variable in per cent divided by 100.
c = control point in per cent divided by 100.
t = time in minutes.

The negative sign is included in equation 4–1 and subsequent controller equations in order to depict the counteraction caused in automatic control. The control valve action must always produce a change in the controlled variable opposite to that which caused the first corrective action. The floating rate is defined as the per cent per minute valve motion caused by 1 per cent deviation of the controlled variable.

Let us suppose that the deviation $(\theta - c)$ is constant at 0.03 in the units above. If the floating rate is set at 20, the rate of valve motion becomes

$$-\frac{dP}{dt} = 20 \times 0.03 = 60 \text{ per cent per minute}$$

or the valve will move at a rate of 60 per cent of its travel in 1 minute.

The controller equation may also be shown in its integrated form in which valve position depends upon deviation and elapsed time of deviation. Thus,

$$-P = f \int_0^t (\theta - c)\, dt + K \qquad [4\text{–}2]$$

where K is a constant of integration determined by the initial valve position. The floating-rate adjustment is similar to an integration factor and merely selects the magnitude of controller response when related to a certain area of deviation.

Proportional-speed floating control is nearly always obtained by hydraulic means. The piston-type hydraulic controller in which the rate of oil supply to the cylinder is adjusted in proportion to the magnitude of the measured variable provides proportional-speed floating action. Sometimes an electric motor is operated from an electrical interrupter in such a manner that power is supplied for various fractions of a cycle. The controller then adjusts the length of the fractions of the cycle in accordance with the deviation of the controlled variable and proportional-speed floating action is obtained.

The application of the proportional-speed floating controller is generally such that fast motion of the final control element is required. The controller lag should, therefore, be as small as possible in order that the valve may follow consistently the rates of motion determined by the magnitude of deviation. Proportional-speed floating controllers of the hydraulic type usually have small controller lag, and this factor becomes correspondingly less important. The dead zone of controllers of these types is generally very small, sometimes measured in hundredths of 1 per cent of controller scale.

Proportional Mode

The proportional, throttling, or modulating mode is a type of controller action in which the final control element is positioned in proportion to the magnitude of the controlled variable. Thus the action of the controlled variable is simply repeated and amplified in the valve action. For purposes of flexibility an adjustment of the mode is usually provided, termed proportional band or throttling range.

The proportional band of a proportional controller is the percentage of full scale change of the controlled variable required to position the valve or other control element from one limit to the other in its operating range. With a narrow proportional band only a small change in the controlled variable is required to operate the valve through its

full range. A large change in the controlled variable is required to move the valve between open and closed positions when the proportional band is wide. Figure 4-12 represents the relationship between controlled variable and valve position for various proportional-band
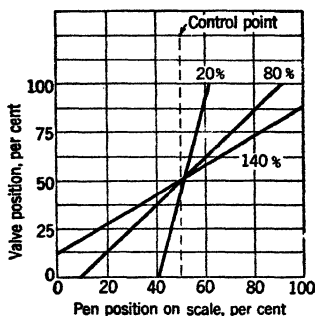
FIG. 4-12. Effect of Proportional-Band Adjustment for a Proportional Controller.
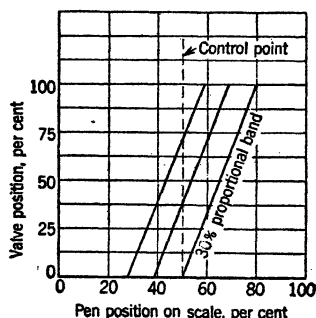
FIG. 4-13. Effect of Manual-Reset Adjustment for a Proportional Controller.

settings. Notice that when the proportional band is more than 100 per cent it is impossible to open or close the valve completely even if the controlled variable should move to zero or full scale of the controller.

Manual reset is used for varying the amount of valve opening when the controlled variable is at the control point. In Fig. 4-12 the valve is 50 per cent open when the variable is at the control point. This requirement may vary, and it may be necessary to hold almost any valve opening for a particular setting of the control point. Figure 4-13 illustrates how the manual-reset adjustment shifts the proportional band along the controller scale in order to obtain 0, 38, and 73 per cent valve openings when the pen is at the control point.

FIG. 4-14. Response of a Proportional Controller to a Sudden and a Stable Change.

In order to analyze the dynamic action of the proportional mode,

the valve movement resulting from a number of different changes in the controlled variable may be drawn. The response of a proportional controller and the effect of the proportional-band adjustment may be noted from Fig. 4–14.

The controller equation for the proportional mode may be developed from Figs. 4–12 and 4–13. It may be expressed in two forms: one relating valve position to controlled variable, and the other relating the rate of valve movement to the rate of change of the controlled variable. Valve position may be shown, assuming a linear controller scale, by

$$-P = \frac{1}{s}(\theta - c) + M \qquad [4\text{–}3]$$

where P = valve position per cent divided by 100.
s = proportional band in per cent divided by 100.
M = a constant depending upon manual-reset position.
θ = variable per cent divided by 100.
c = control point per cent divided by 100.

Controllers of the proportional type are generally linear when pen position in inches is related to valve position. However, many controller scales, such as those for the vapor-actuated thermometer, radiation pyrometer, and differential pressure flowmeter, are not linear, and equation 4–3 must be corrected.

It is interesting to solve this equation for a few particular conditions. Let us suppose that the pen is at the control point. Therefore,

$$(\theta - c) = 0$$

Then, since the valve may be in a 50 per cent position as shown in Fig. 4–13, the constant M becomes 0.50 for manual reset, and equation 4–3 becomes

$$-P = \frac{1}{s}(\theta - c) + 0.50$$

Now assume that the proportional band is 20 per cent and that the pen departs from the control point by 7 per cent. Then equation 4–3 becomes

$$-P = \frac{1}{0.20}0.07 + 0.50$$

or

$$-P = 0.85$$

Under these circumstances the valve would have moved from 50 to 85 per cent open.

Another way of expressing the same relationship is by differentiating both sides of equation 4-3. Thus,

$$-\frac{dP}{dt} = \frac{1}{s}\frac{d\theta}{dt} \quad [4\text{-}4]$$

This relationship shows that the rate of valve movement is proportional to the rate of change of the controlled variable. We may solve this equation for a particular example by assuming a proportional band of 20 per cent and a rate of temperature change of 7 per cent per minute. Then

$$-\frac{dP}{dt} = \frac{1}{0.20}0.07$$

or

$$-\frac{dP}{dt} = 0.35$$

The rate of valve movement, then, would be 35 per cent per minute when the temperature changes at a rate of 7 per cent per minute.

By means of this relationship, it is possible to determine the maximum speed of valve movement required because, in the example above, if the maximum rate of temperature change were 7 per cent per minute, then the valve would be required to move at a speed of 35 per cent per minute, or to move full travel in at least 2.86 minutes.

Pneumatic Proportional Controller

A pneumatic controller may be operated by either a self-operated or power-operated measuring means since but small power is required for its operation. Pressure thermometers, flowmeters, pressure gages, and self-balancing potentiometers with pneumatic control are very common. The operation of a proportional controller is based on the principle of modulating the response of a two-position controller. A pneumatic controller illustrates how this is done.

The arrangement of a two-position controller is shown schematically in Fig. 4-15. The pressure transmitted through the line to the control valve depends upon a balance between the supply and leakage of air. A baffle or flapper positioned by the measuring means controls the leakage of air from the nozzle. When the pen is above the control point the flapper covers the nozzle opening, the pressure increases, and the control valve closes. When the pen is below the

control point the flapper uncovers the nozzle and the control valve opens.

The minimum proportional band of the pneumatic on-off controller is generally between 0.0 and 0.5 per cent of controller scale. It is possible, then, for the control valve to assume some opening between full open and full closed when the controlled variable is within a fraction of 1 per cent of the control point.
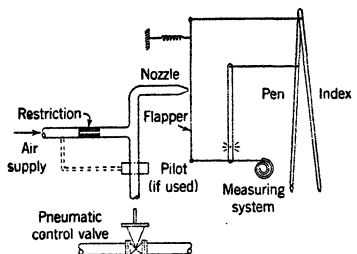
FIG. 4–15. Schematic Diagram for a Pneumatic Two-Position (On-Off) Controller.

The restriction in the supply line determines the amount of air supplied to the system. Thus, for the pressure in the system to change from minimum to maximum the full volume of air required to build up the pressure must pass through the restriction. In order to decrease the pressure all the air must leak out through the nozzle.

If the output volume, consisting of the volume of the diaphragm valve top and the connecting line, is large, the rate of pressure change may be quite slow. For this reason an amplifying pneumatic relay or pilot is often added to the flapper and nozzle in order to increase the capacity of the air system and to decrease the lag of changing the pressure. The pilot has a separate air supply and is controlled by the nozzle back-pressure. The controller lag with the on-off pneumatic controller will be considered with proportional controllers.

The pneumatic proportional controller is shown in the schematic drawing of Fig. 4–16. The only change from Fig. 4–15 is the addition of the feedback bellows, to which the output pressure of the pilot is connected. A pilot is nearly always employed in pneumatic proportional controllers. When the pen rises a small amount the flapper covers the nozzle and the output pressure from the pilot increases. The increase in output pressure operates through the feedback bellows to move the flapper in the opposite direction from its first motion and stabilize the output pressure at a slightly increased value. Negative feedback or follow-up action is also employed in
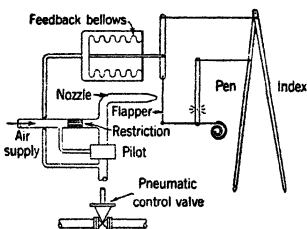
FIG. 4–16. Schematic Diagram for a Pneumatic Proportional Controller.

electronic amplifiers and potentiometer balancing systems in order to obtain stable operation.

The pneumatic transmission system, mentioned in a previous chapter, is constructed in a manner identical with the pneumatic proportional controller. In a transmission system a fixed proportional band of approximately 100 per cent provides a relation between measured variable and output pressure.

Fig. 4–17. Response Curves for a Typical Pneumatic Controller.

The dead zone of a pneumatic controller or transmission system is extremely small, and definite changes in output pressure are generally obtained with less than 0.001 per cent of full scale change in pen position. The dead zone of a complete pneumatic controlling means is somewhat greater when a control valve is included. With a normal amount of valve friction and a moderate proportional band, the dead zone should be less than 0.15 per cent. A valve positioner decreases the dead zone to about 0.05 per cent.

The controller lag of a pneumatic controller may be expressed in terms of minutes lag coefficient, similar to measuring lag. Figure 4–17 shows the response of a pneumatic controller to a sudden change in controller pen position. The lag of the controller response depends, to a large extent, upon the capacity of the pilot and the volume of the controller output connection.

Although the responses shown in Fig. 4–17 are not quite logarithmic, we may approximate the controller lag by measuring the time to attain 63 per cent of the complete change. The table below shows the extent of controller lag for a pneumatic controller:

	Controller Lag Minutes
50-ft tubing plus small volume	0.007
200-ft tubing plus small volume	0.02
50-ft tubing plus large valve	0.05
200-ft tubing plus large valve	0.14

The controller lag is greatly increased by larger output capacity but generally is not affected to any great extent by longer lengths of tubing.

The connecting line or tubing should be neither too large nor too small. An optimum value for most controllers is about ¼ inch internal diameter.

With a very short length of connecting line and a small end volume, a pneumatic controller may oscillate slightly because the output volume is insufficient to damp the controller action. Under these circumstances it is advisable to add volume. This volume is much more effective in the form of tubing; 20 to 40 feet extra length is usually sufficient. The addition of a volume tank substantially increases the controller lag.

Proportional-band and manual-reset adjustments are provided in most pneumatic proportional controllers by modifying the linkage between the controller pen, flapper, and feedback motions.

Electric Proportional Controller

The industrial type of electric proportional controller is generally limited to use with temperature-measuring means of the automatically balancing potentiometer type because the amount of power required by the controlling means is ordinarily greater than that available in self-operated means such as pressure thermometers or millivoltmeters.
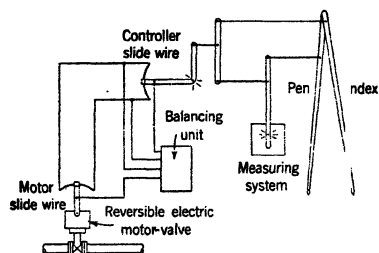
Fig. 4–18. Schematic Diagram for a Bridge-Type Electric Proportional Controller.

One type of electric proportional controller operates on the principle of modulating the response of an on-off control system by means of a Wheatstone bridge, and balancing the motor valve position against the position of the pen. Figure 4–18 illustrates one such arrangement.

When the instrument pen rises a small amount the resistance of one side of the bridge becomes greater than the other. This causes a small current to flow between the controller and motor slide-wire contactors. The controller balancing unit detects the unbalance and drives the valve motor in a direction to rebalance the bridge. This operation provides a given motor position for each deviation of the pen from the control point.

The controller balancing unit may be operated by electric or electronic means or by a combination of the two. The potentiometer controller, then, has two balancing arrangements, one for the meas-

uring means and one for the controlling means. One arrangement of the electric proportional controller switches a balancing unit periodically from the measuring circuit to the control circuit.

The dead zone and controller lag for the electric bridge-type proportional controller are interrelated since with no dead zone the motor valve would follow temperature changes with negligible lag. The speed of operation of the electric motor valve is generally between 0.25 and 1.0 minute for full stroke. Therefore, as long as the temperature does not change rapidly, little controller lag is present when the dead zone of operation is zero.

The dead zone of the electric bridge-type proportional controller is generally about 0.15 to 0.6 per cent of controller scale, assuming a moderate proportional band. Consequently, any low rate of movement is approximated by a series of short steps. The motor valve then lags behind the proper position by a time proportional to the dead zone. Therefore, a large dead zone brings about a large controller lag.

The advantage of the electric proportional controller is that the motor valve may be located almost any reasonable distance from the controller. The controller lag is not appreciably increased by longer connecting lines between the valve and controller slide wires.

A second type of electric proportional controller for electrical furnaces adjusts a saturable-core reactor, which varies the current flow to the heating electrodes of the furnace. The current flow to the furnace is made proportional to the temperature of the furnace, thereby accomplishing proportional control.

A third type of electric proportional controller for electrical furnaces interrupts the current flow to the furnace for various fractions of a cycle. If the length of the fractions of the cycle are adjusted in response to magnitude of furnace temperature, proportional control results through the average rate of current flow.

Operation of Reset Response

Proportional-speed floating controller action is often combined additively with proportional controller action, the combination being termed a proportional-reset mode of control.

We observed in the previous section that a manual adjustment on a proportional controller is necessary in order to provide for different valve openings when the variable is at the control point. The manual adjustment may be made automatic by resetting the controlling action (but not the control index) of the controller. In other words, reset response simply shifts the proportional band up or down the scale of a

controller until the required valve opening returns the variable to the control point.

This operation may be visualized by the method shown in Fig. 4–19. The vertical line represents the control point, and the shaded portion represents the proportional band. The amount of valve opening is shown by the position of the temperature in the proportional band.
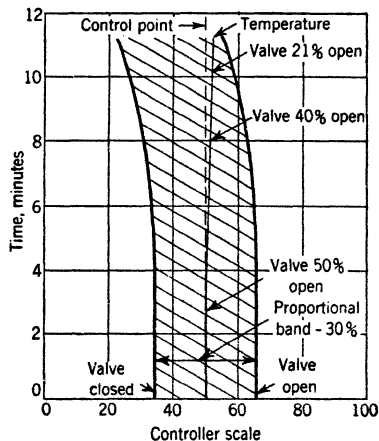
FIG. 4–19. Characteristic of Reset Response in Shifting Proportional Band on Controller Scale.
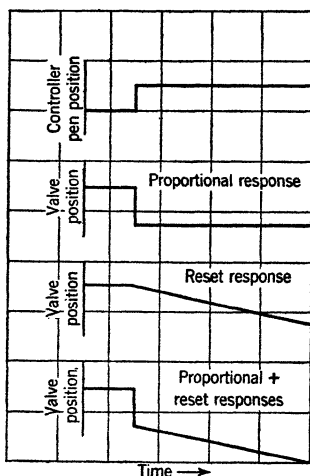
FIG. 4–20. Response of a Proportional-Reset Controller to a Sudden Change.

Thus, at the beginning, the temperature is at the control point and the valve is half open, but, as the temperature begins to depart from the control point, reset response action shifts the proportional band in the opposite direction so as to provide a further decrease in valve opening. This action ultimately forces the variable toward the control point.

It will be instructive to select a specific change in the controlled variable and analyze the resulting response of the controller. In Fig. 4–20 a small instantaneous movement of the variable is assumed. The controller lag is neglected in order to illustrate clearly the control responses. The opposite directions of change of the variable and control responses depict the opposing action of control. The proportional response duplicates the nature of the change of the controlled variable. Reset response gradually changes the valve setting. Since the variable has a fixed departure from the control point, reset response

moves the valve at a constant rate, as previously shown in Fig. 4–10.

The algebraic sum of the two responses is the action of the proportional-reset mode. The valve or other final element quickly closes a slight amount and then continues to close more gradually. If the pen stayed at the new deviation long enough, the valve would continue in that direction until it became entirely closed.
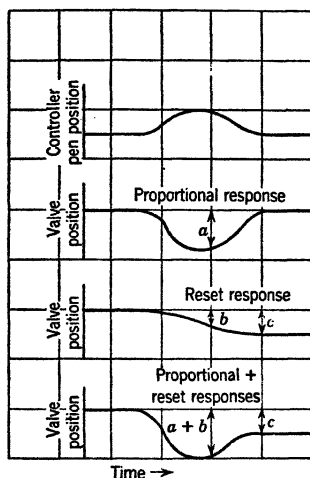
Fig. 4–21. Response of a Proportional-Reset Controller to a Stable Change.
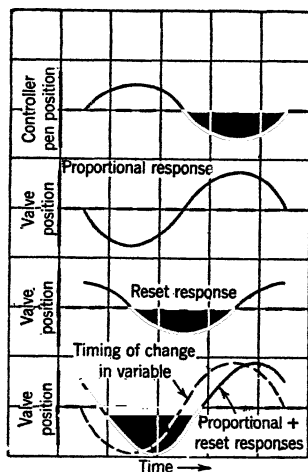
Fig. 4–22. Response of a Proportional-Reset Controller to a Cycling Variable.

Since the change of the variable in Fig. 4–20 is more for the purpose of illustration than for practical application, another change is assumed in Fig. 4–21. The proportional response duplicates the change in the variable as in Fig. 4–14. The reset response continuously integrates the area under the deviation curve as in Fig. 4–11.

The algebraic sum of the responses is the action of a proportional-reset controller. The valve first closes a large amount at a fast rate as the variable deviates from the control point and then opens a small amount at a slower rate as the variable returns to the control point. The valve is finally left in a position slightly more closed than at the beginning of the change.

Figure 4–22 illustrates a cycling temperature which we will assume is a sine wave. As before, the proportional response repeats the nature of the change of the variable. The reset response, however, is of different form, since the integral of a sine wave is a positive cosine

wave. When the responses are added together, a wave of the same type (sine or cosine) is produced, but the addition of the reset action has changed the phase, or, in other words, produced a displacement, of the response curve. In order to illustrate this displacement, the variable has been inverted, slightly amplified, and plotted with the controller-response curve so as to show the relative timing of the response. The solid curve is the valve response, and the dotted curve is the change of the variable. The reset response has, by its own nature, caused a lag in valve response.

FIG. 4-23. Effect of Reset Rate Adjustment for a Proportional-Reset Controller.

The proportional-reset controller has two adjustments: a proportional-band adjustment, as described in the previous section, and a reset-rate adjustment. The reset-rate adjustment fixes the relative magnitude of the proportional-speed floating action. In Fig. 4-23 the reset response to a constant pen deviation and the effect of various reset adjustments are shown. A slow reset rate is one having a small rate of floating, and a fast reset rate is one having a large rate of floating.

The controller equation for the proportional-reset mode is the sum of the two individual responses. Assuming a linear controller scale,

$$-P = \frac{r}{s}\int_0^t (\theta - c)\,dt + \frac{1}{s}(\theta - c) + K \qquad [4\text{-}5]$$

where P = valve position, per cent divided by 100.
r = reset rate, units per minute.
s = proportional band, per cent divided by 100.
θ = variable, per cent divided by 100.
c = control point, per cent divided by 100.
t = time, minutes.
K = constant of integration, initial valve position.

The units of reset rate r are the inverse of minutes or a number per minute. Reset rate is defined, then, as the number of times per minute

that the proportional response is duplicated when the controlled variable suddenly changes. In Fig. 4–23 the reset rates are about 0.04 for the slowest, 0.14 for the medium, and 0.33 for the fastest rate.

Let us suppose that the deviation of the variable $(\theta - c)$ is a constant at 0.03. Equation 4–5 becomes, then,

$$-P = \frac{0.03r}{s} \int_0^t dt + \frac{0.03}{s} + K$$

Now, if the reset rate r is set at 0.05 per minute and the proportional band is 0.20,

$$-P = \frac{0.03 \times 0.05}{0.20} t + \frac{0.03}{0.20} + K$$

The valve position at any time would be given by

$$-P = 0.0075t + 0.15 + K$$

and at the end of 1 minute the valve position would be

$$-P = 0.1575$$

or about 16 per cent open, provided that the valve started originally from zero so that the constant of integration of the above equation is zero.

Equation 4–5 is often stated in another manner in order to express the rate of valve movement rather than the valve position. The rate of valve movement is obtained by differentiating the equation above. So,

$$-\frac{dP}{dt} = \frac{r}{s}(\theta - c) + \frac{1}{s}\frac{d\theta}{dt} \qquad [4\text{–}6]$$

We may determine the rate of valve movement for the example above by substituting the assumed quantities. Thus,

$$-\frac{dP}{dt} = \frac{0.05}{0.20} \times 0.03 + \frac{1}{0.20} \times 0$$

or

$$-\frac{dP}{dt} = 0.0075$$

This shows that the valve would be moving because of reset action alone at a rate of 0.75 per cent of travel per minute.

The magnitude of reset response depends not only upon the reset-rate adjustment but also upon the proportional-band setting. In the example above, with a proportional band of 20 per cent, the resulting

rate of valve motion was 0.75 per cent of valve travel per minute. If the proportional band is set to 10 per cent, the rate of valve motion becomes

$$-\frac{dP}{dt} = \frac{0.05}{0.10} \times 0.03 = 0.015$$

or 1.5 per cent per minute. Thus, if the proportional band is halved, the reset response is doubled, even though the reset-rate adjustment is unchanged. Controller adjustment thereby is actually made more versatile and simple even though the interaction may seem complicated.

The mechanisms for proportional-reset controllers are not simple; rather than give an inadequate description that might be confusing, it is recommended that other sources be consulted. Manufacturers' catalogs are especially suitable.

Operation of Rate Response

Rate response is sometimes added to the proportional or to the proportional-reset controller. Other terms for this mode of control are: derivative response, lead component, and rate component. Rate response may be defined as a control response in which the valve position may be proportional to the rate of change of deviation.

Reversing the usual procedure, it will simplify the discussion to consider the controller equation first rather than last. From the definition above, it will be seen that another term is added to equation 4–5 for the proportional-reset controller.

$$-P = \frac{r}{s}\int_0^t (\theta - c)\,dt + \frac{1}{s}(\theta - c) + \frac{q}{s}\frac{d\theta}{dt} + K \qquad [4\text{–}7]$$

This equation illustrates the effect of the *rate of change of variable* on *valve position* and suggests the definition given above. As before, the equation may be differentiated,

$$-\frac{dP}{dt} = \frac{r}{s}(\theta - c) + \frac{1}{s}\frac{d\theta}{dt} + \frac{q}{s}\frac{d^2\theta}{dt^2} \qquad [4\text{–}8]$$

Expressing the controller equation in this form reveals a second definition for rate response whereby *rate of valve movement* responds, in part, to the second derivative or *rate of the rate of change of variable.* These two definitions are discussed in detail so that confusion will be avoided.

The units of the rate-time adjustment q are the inverse of the units of reset rate, that is, minutes. The physical interpretation of the constant q may be found by setting r equal to zero so that there is no

reset response and setting the variable to cross the control point at a rate of 6 per cent per minute. If the proportional band is 20 per cent, then

$$(\theta - c) = 0 \qquad \frac{d\theta}{dt} = 0.06$$

$$s = 0.20 \qquad r = 0$$

and, substituting in equation 4–7,

$$-P = \frac{1}{0.20} \times 0 + \frac{q}{0.20} \times 0.06 + K$$

$$-P = 0.30q + K$$

The action described by this example is illustrated in Fig. 4–24 in order to demonstrate the meaning of rate time. The variable crosses the control point at the rate of 6 per cent of controller scale per minute. The proportional response duplicates this action. Rate response provides a constant increment of valve motion since the rate of change of the variable is constant. Adding the responses together algebraically, the response of the proportional-rate mode advances the valve position with respect to time. For a rate time of 0.25 minute the valve action is advanced 0.25 minute, and for a rate time of 1.0 minute the valve action is advanced 1.0 minute.

Fig. 4–24. Effect of Rate-Response Adjustment for a Proportional-Rate Controller.

The magnitude of rate-response action is also dependent upon the adjustment of proportional band. In the sample calculation above, a rate-time setting of 0.25 minute results in an additional increment of valve position of

$$-P = 0.30 \times 0.25 = 0.075$$

or 7.50 per cent of valve travel. Suppose, however, that the proportional band is set to one-half its previous value. The rate-response

action then produces an additional increment of

$$-P = \frac{q}{0.10} \times 0.06 = 0.60 \times 0.25 = 0.15$$

or 15 per cent of valve travel. This dependence of the rate-response action upon both the rate-time and proportional-band adjustments makes for easier adjustment of controller responses.

If the controlled variable were to change as in Fig. 4–25, the proportional response would repeat the action while the reset response
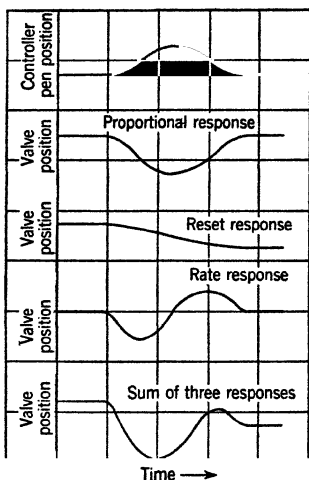
Fig. 4–25. Response of Proportional-Reset-Rate Controller to a Stable Change.
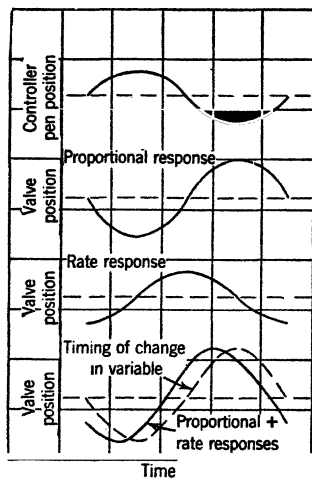
Fig. 4–26. Response of a Proportional-Rate Controller to a Cycling Variable.

would integrate the area under the curve. The rate of change or slope of the variable is at first zero, increases to a maximum, returns to zero, becomes negative, and again returns to zero. It should be noted that when the controlled variable is returning to the control point the valve has almost returned to its original position before settling at the final opening because the rate of change of the variable at that point is negative. The faster valve movement at the beginning of the change and the subsequent cycling of the valve is caused by the addition of rate response.

When rate response is used with the controller and the controlled variable is cycling, the result is opposite to the operation of reset re-

sponse. Suppose in this example that the reset rate r is zero so that the effect of rate response can be analyzed separately. Figure 4–26 shows the curves for the controlled variable and for the proportional response. However, rate response reacts to the slope of the curve for the change in controlled variable and differentiates the proportional curve, giving a negative cosine wave.

The sum of the proportional and rate responses is shown along with the curve of the controlled variable — inverted and amplified so as to indicate time relationship. The effect of rate response is to advance the valve action. By constructing another figure similar to Fig. 4–26, and including the response due to reset, it is possible to show that a proper amount of rate response will compensate for the inherent lag of reset response at one given period.

Rate response may or may not have a separate adjustment. Equation 4–8 was

$$-\frac{dP}{dt} = \frac{r}{s}(\theta - c) + \frac{1}{s}\frac{d\theta}{dt} + \frac{q}{s}\frac{d^2\theta}{dt^2} \qquad [4\text{–}8]$$

where s = proportional band, adjustable, affects all terms, in per cent divided by 100.
r = reset rate, adjustable, in units per minute.
q = rate time, may be adjustable, in minutes.

The rate time q may be fixed or variable, depending upon the arrangement of the controller mechanism. If the adjustment is fixed, an adjustment of the proportional response automatically varies the magnitude of rate response because both q and s are included in the rate term. On the other hand, a lack of adjustment in the rate time q fixes the relationship between the proportional and rate responses.

The use of rate response requires an ability in the controller to respond rapidly because an appreciable controller lag may nullify the advantages gained in using rate response. The controller mechanism must have a maximum rate of movement of almost twice as much as when proportional and reset responses alone are used since rate response is employed for the purpose of providing large, sudden movement of the valve. Although rate response will aid in overcoming the effect of controller lag, a large controller lag will make response to rapid rates of change of the variable impossible.

Theory of Continuous Controllers

We have noticed a symmetry in the controller equations for proportional controllers. This symmetry is logical and may be shown

by the following general equation for controller operation:

$$-\frac{dP}{dt} = K_a \int_0^t \theta\, dt + K_0\theta + K_1\theta' + K_2\theta'' + K_3\theta''' \cdots K_n\theta^n \quad [4\text{–}9]$$

where $\frac{dP}{dt}$ = rate of valve movement.

θ = deviation of variable (the primes indicate the first, second, third derivative, and so on).

K = an adjustable constant for each response.

The terms each indicate a response so that

$K_0\theta$ = reset response

$K_1\theta'$ = proportional response

$K_2\theta''$ = rate response

and these three responses are the ones in general use.

A continuous controller is one which includes one or more of the terms of equation 4–9 but each response must be made proportional to the controlled variable or one of its derivatives. For example, the proportional-speed floating controller includes only the second term of the equation, thus:

$$-\frac{dP}{dt} = f\theta \quad [4\text{–}10]$$

The proportional-speed floating controller can be successfully applied to many processes and the controlled variable will be stable because when the deviation θ is zero the valve will remain stationary.

The single-speed floating controller, however, does not respond to the magnitude of the controlled variable, and this mode of control is not continuous.

$$-\frac{dP}{dt} = K_1 \frac{\theta}{|\theta|} \quad [4\text{–}11]$$

With this mode of control, however, the valve motion cannot be stopped when the variable is at the control point since the valve motion is not responsive to the magnitude of deviation. For this reason a neutral is required in order to hold the valve stationary when the deviation is near zero.

Single-speed and proportional-speed floating controllers must therefore be carefully distinguished. Either mode is employed as reset response in conjunction with the proportional mode in different types

of proportional controllers. The pneumatic or hydraulic proportional-reset controllers nearly always use the proportional-speed floating mode for reset response.

As indicated by equation 4–9, it would be possible to add the third, fourth, or higher derivatives, or even an integral term, to the response of the continuous controller. From studying the mechanisms of continuous controllers it will be found that in general an extra piece of equipment must be added to obtain a response to each additional derivative. A controller mechanism to include four terms of the equation would include four coordinated units. Therefore, a limit of a physical and economic nature is soon reached where added derivatives become useless.

Each additional derivative in the controller response requires adjustment. The total number of adjustments need not be equal to the exact number of derivatives, but there should be at least two. The setting of three, four, or five semi-independent adjustments is often prohibitively difficult in practice.

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CHAPTER 5

FINAL CONTROL ELEMENTS

There is probably no other unit in the control system which receives so little consideration as the valve or final control element. In many systems a control valve is subjected to more severe conditions of temperature, pressure, weather, and dirt than the primary element, and yet it must perform satisfactorily with a minimum of attention.

The final control element is the mechanism which varies the flow of control agent in response to a signal from the controller. Although it is usually a valve, the final element may be a damper, a louver, a pump, a motor, or any such unit, depending upon the method by which control is accomplished. As with controllers, the final control element may be operated by pneumatic, electric, hydraulic, or mechanical means.

The control agent determines in a large measure the selection of the control valve or other element. The control agent may be any fluid, such as air, fuel gas, fuel oil, hydrocarbon, refrigerant, or acid. The temperature may range from below zero to above 1000° F; the fluid may be either alkaline or acid; the pressure may be very low or very high.

The operating characteristic of the controlling means, that is, the relationship between the change in controlled variable and the corresponding change in flow of the control agent, depends to a great extent on the flow characteristic of the final control element. With controllers of the continuous type it is very important that smooth and even changes are made in the flow to the process.

The varieties and types of final control elements are numerous, but a description of pneumatically and electrically operated valves, dampers, and louvers will serve to clarify their operating characteristics and principles of operation. Electrical relays often serve as final control elements, particularly for two-position (on-off) electric control of electric furnaces.

FLOW EQUATIONS FOR VALVES

Since the final element generally controls the flow of a fluid it acts as a variable orifice or restriction. The laws of fluid mechanics are applicable, particularly the formula for streamline flow through an orifice:

$$F = KA\sqrt{2gh} \qquad [5\text{-}1]$$

where F = flow in cubic feet per second.
K = constant.
A = area in square feet.
g = acceleration due to gravity.
h = pressure head in feet of fluid flowing.

This equation shows that the flow is directly proportional to the area and proportional to the square root of the pressure head. However, the final element is actually a submerged orifice over which a pressure differential exists, and

$$F = KA\sqrt{2g(h_1 - h_2)} \qquad [5\text{-}2]$$

where h_1 = pressure head in feet upstream.
h_2 = pressure head in feet downstream.

For the *flow of liquids* equation 5–2 must be further altered because of the different physical characteristics of a fluid. Specific gravity must be considered in converting pressure head into pressure, and

$$F = K_L A \sqrt{\frac{p_1 - p_2}{G}} \qquad [5\text{-}3]$$

where F = flow in cubic feet per minute.
A = area in square inches.
p_1 = upstream pressure in pounds per square inch absolute.
p_2 = downstream pressure in pounds per square inch absolute.
G = specific gravity of fluid on upstream side.
K_L = constant.

In one particular valve, constant K_L is 1.53 on the assumption that area A is calculated from the nominal pipe diameter of the valve. For example, if the nominal valve size is 2 in., the upstream pressure 100 lb per sq in. gage, and the downstream pressure 90 lb per sq in. gage, the flow of water at full open valve setting would be

$$F = 1.53\,\frac{\pi}{4} \times 2^2 \sqrt{\frac{114.7 - 104.7}{1}}$$

$$F = 15.1 \text{ cu ft per min}$$

For the *flow of gas* an adiabatic change of the gas flowing from the high-pressure region to the low-pressure region is ordinarily assumed. A gas is compressible, and in flowing through an orifice it changes in volume and therefore in internal energy; a practical equation must take

these effects into account. The proper equation is

$$F = K_G A \sqrt{\frac{(p_1 - p_2)p_2}{G}} \qquad [5\text{–}4]$$

The units are the same as in equation 5–3. In one particular valve, constant K_G is 11.5 on the assumption that area A is calculated from the nominal pipe diameter of the valve.

For the *flow of saturated vapor* through an orifice, there is a change of internal energy as well as a change of state in the vapor. These factors being taken into account, the relationship for the flow of a saturated vapor such as steam is

$$F = K_s A V \sqrt{(p_1 - p_2)p_2} \qquad [5\text{–}5]$$

where V is the specific volume in cubic feet per pound. For a particular valve, constant K_s is 0.61 for steam on the assumption that area A is calculated from the nominal pipe diameter of the valve.

The relationships given by equations 5–3, 5–4, and 5–5 are used in estimating valve sizes. They do not apply to the characteristic of flow at any point other than maximun opening. The changes in coefficient K with velocity, pressure differential, and fluid characteristics limit the application of such empirical relationships.

Velocity and Pressure Differential

It will be noticed in the above equations that if the flow is divided by the area the same equation also expresses velocity of flow in the valve. In the sample calculation above, the velocity would be approximately

$$\frac{F}{A} = \frac{15.1 \times 144 \times 4}{\pi \times 2^2} = 692 \text{ ft per min}$$

Velocity, therefore, plays no part in the selection of valve sizes inasmuch as it is determined solely by the pressure differential and the valve opening.

It is assumed that the flow of liquids is laminar, i.e., a smooth flow without turbulence. This condition is rarely encountered in actual practice because the velocity may be higher than the value corresponding to the critical Reynolds' number and the irregularities inside the valve body and in the piping system break up the streamline flow. Turbulent flow causes an increased pressure loss because of the energy lost in turbulence. The flow therefore decreases proportionately as the turbulence due to high velocity increases.

For the flow of gases the velocity is limited in any event by a velocity equal to the acoustic velocity. At this point no further increase in flow

can be obtained because the gas is moving at a velocity as great as the speed at which a pressure wave can be transmitted through the pipe under the existing conditions. A further increase in either area or pressure differential will not increase the velocity or the rate of flow. It is important that the velocity does not approach this value in any installation because, if it did, the valve would not be able to control the flow.

The pressure loss through the lines leading to and from the valve must be taken into account, for it must be subtracted from the total pressure differential available in order to obtain the actual pressure differential at the valve. The effect of non-uniformities in the line, such as fittings, valves, and bends, is to cause a pressure drop since the velocity distribution in the pipe is disturbed. The result is a loss of kinetic energy which is not recoverable, and a part of the pressure head is lost.

This pressure loss in lines and fittings varies as the square of the fluid velocity in accordance with the basic-flow relationship stated in equation 5–1. It is advisable, therefore, to have a relatively low velocity of flow in the lines leading to and from the control valve in order to keep the variation in pressure differential at a minimum. In general, the velocity of a liquid in the lines leading to and from the valve should be less than 300 ft per min. The velocity of a gas may vary considerably, depending upon its characteristics.

When a valve or other final control element is installed, a pressure differential must exist across the element in order to obtain a flow. This pressure differential is not an arbitrarily selected value but depends upon the arrangement of the piping system.
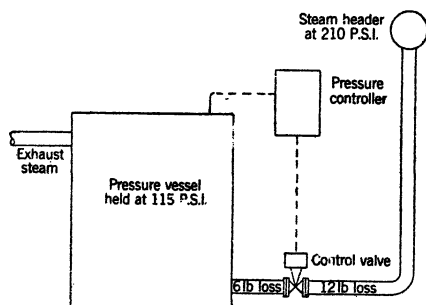
FIG. 5–1. Pressure Differential at a Control Valve.

Let us suppose that the pressure controller of Fig. 5–1 is holding the pressure at the vessel constant at 115 lb per sq in. by controlling the flow of steam through a control valve. The header pressure will be assumed as 210 lb per sq in. Suppose further that the line losses from header to valve are 12 lb per sq in. and the losses from valve to vessel are 6 lb per sq in. The upstream pressure at the valve will be $210 - 12 = 198$ lb per sq in. The pressure differential at the valve is *not any preselected value* but must be $198 - 121 = 77$ lb per sq in.

The total pressure differential available includes both the line losses

and the pressure differential at the valve. If the line losses are large the pressure differential across the control valve must be small by comparison. For example, it would be possible to increase the velocity of flow to such an extent that the lines and fittings would absorb all the available pressure differential. The valve would then become completely ineffective in controlling the flow.

The loss of control is shown by Fig. 5–2. The curve $K = 1$ represents the flow characteristic of the valve or other control element. This flow characteristic is obtained by setting the valve at various openings and measuring the flow. It is representative of the action of the control system, since the controller regulates the valve opening, thereby adjusting the flow to the process.
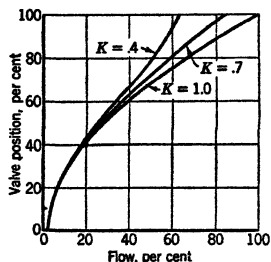
FIG. 5–2. Effect of Variation in Line Losses.

The curves of Fig. 5–2 represent the flow characteristic when K is the ratio of the pressure differential existing at the valve to the total differential available. The maximum flow through the valve is seriously affected by increases in line velocity. Notice that, if the velocity should reach a point where only 40 per cent of the differential is at the valve ($K = 0.40$), the valve would pass only about 63 per cent of its maximum flow. Line losses greater than 50 per cent of the available differential would practically put the controller out of operation because the normally required flow could not be obtained.

The pressure differential at a final element should be at least 60 per cent and preferably 70 per cent of the available differential. Line losses must be minimized in order to provide the maximum differential at the valve. The maximum flow as well as the flow characteristic can thereby be maintained and the final element will provide effective control of the flow of the control agent.

RANGEABILITY AND TURNDOWN

The curve of Fig. 5–3 is the area characteristic of the valve; it is determined by plotting the area of the valve at various openings against valve position. According to equation 5–1 the area is proportional to the flow when the pressure differential remains constant. In studying flow characteristics, the area characteristic is often used since the flow characteristic varies considerably with installation and is therefore unknown unless actually measured. It must be remembered, however, that the pressure differential is not constant in any installation and the

actual flow characteristic may be substantially different from the area characteristic.

The area characteristic of Fig. 5–3 is semi-logarithmic — a characteristic which provides *equal percentage* increases in area for *equal increments* of valve movement. An equal percentage characteristic fulfills the requirements of the following equation:

$$P = K \log_e \frac{F}{F_0} \qquad [5\text{–}6]$$

where F = flow in per cent of maximum.
F_0 = constant.
K = constant.
P = valve position in per cent of maximum.

When the valve moves to a minimum position the flow does not become
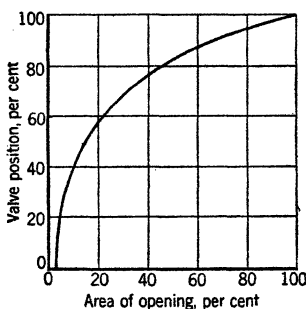
FIG. 5–3. Semi-Logarithmic Area Characteristic on Rectangular Coordinates.
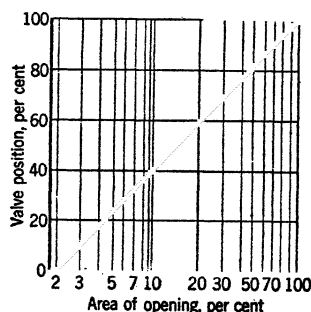
FIG. 5–4. Semi-Logarithmic Area Characteristic on Semi-Logarithmic Coordinates.

zero but a minimum of F_0. The constant F_0, then, represents the minimum controllable flow for the particular design of final element. This is shown by solving equation 5–6 as follows:

$$0 = K \log_e \frac{F}{F_0}$$

and

$$F = F_0$$

If a semi-logarithmic characteristic is plotted on semi-log coordinates, a straight line will be obtained as in Fig. 5–4. We may now determine the characteristic equation for this valve because F_0 here equals 2.0 per cent of maximum flow. Thus,

$$P = K \log_e \frac{F}{2}$$

The constant K may be evaluated at the maximum flow point because, when the lift is 100 per cent, the flow is 100 per cent. Thus,

$$100 = K \log_e \frac{100}{2}$$

$$K = \frac{100}{\log_e 50} = 25.6$$

Therefore, the equation for the valve of Figs. 5–3 and 5–4 is

$$P = 25.6 \log_e \frac{F}{2}$$

The minimum controllable flow of a final element is dependent upon its construction. Clearances must be allowed in order to avoid binding and sticking, and the flow through these clearances constitutes a necessary leakage. The minimum controllable flow, or F_0 in the above equation, may vary from 1.50 to 4.00 per cent of maximum flow. Since clearance is more or less constant regardless of valve size, the minimum controllable flow will be a greater percentage of maximum flow in smaller valves.

The rangeability of the final element is determined by the minimum controllable flow. It is defined as the number of times the minimum flow may be increased before maximum flow is obtained, or as the ratio of maximum to minimum controllable flow. Therefore,

$$\text{Rangeability} = \frac{100}{F_0} \qquad [5\text{–}7]$$

For the valve of Fig. 5–4 the rangeability is

$$\frac{100}{2} = 50$$

From the values given above for minimum controllable flow, we see that the rangeability of a valve is generally between 66 and 25.

The turndown of the final element is also determined by the minimum controllable flow but is based upon the normal maximum instead of the maximum flow. It is defined as the number of times the minimum flow may be increased before normal maximum flow is obtained, or as the ratio of normal maximum to minimum controllable flow. Therefore,

$$\text{Turndown} = \frac{\text{Per cent of normal maximum flow}}{F_0}$$

If the normal maximum flow through a valve is 70 per cent and the minimum controllable flow is 3.5 per cent, the turndown is

$$\frac{70}{3.5} = 20$$

Turndown therefore depends to a great extent upon the size of the valve and whether a large or small flow is required as a normal maximum.

The importance of rangeability and turndown lies in its application to the process equipment. For example, if the design of the process equipment requires that the operating flow of the control agent be reduced 30 times in order to maintain a desired balance of conditions within the process, then the turndown must be at least 30, and the rangeability must be at least 44 if a normal operating flow of 70 per cent is assumed.

FIG. 5-5. Typical Installations of a Pneumatic Diaphragm Control Valve.

An arrangement of the control element is sometimes made as in Fig. 5-5 where a portion of the flow of the control agent is bypassed around the control valve by means of a hand valve which is set partly open. For example, one might wish to maintain a small flow of fuel to the burners in a furnace so that the flame would not die out if the control valve should completely close.

The curves of Fig. 5-6 show the flow characteristic for various openings of the bypass valve. Here the ratio K is the ratio of maximum flow through the bypass to maximum flow through the valve. When the bypass is set so that zero flow is bypassed, $K = 0$. One of the effects of a bypass is that the flow characteristic changes with different settings of the bypass. Thus what was originally a semi-logarithmic characteristic without the bypass is substantially altered when the bypass valve is opened.

FIG. 5-6. Flow Characteristic with a Bypass around the Control Valve. (Courtesy of Peters and the American Chemical Society, reference 4.)

The most serious effect of a bypass is the reduction of the rangeability and turndown as shown in the table below calculated from Fig. 5-6. In most applications such a reduction of rangeability would not be permissible. Therefore, a partially open bypass around the valve should always be avoided.

Although the flow should not be bypassed around the control valve the installation shown in Fig. 5-5 is a generally accepted method. The bypass valve should remain closed. This method of installation allows

Per Cent Flow in Bypass	Rangeability	Turndown (assume normal maximum flow of 70 per cent)
0	50	35
6	12	8.8
11	7.7	5.4
20	4.5	3.2
33	3.0	2.0
50	2.0	1.4

the control valve to be removed and permits manual control by means of the bypass valve if it is necessary to service the control valve.

The size of the final control element is important in the operation of the control system because of its effect on rangeability and flow characteristic. The control valve or other element is selected as to size by means of equations 5–3, 5–4, and 5–5. Before solving these equations it is necessary to know the characteristics of the control agent, the pressure differential across the valve, and the flow required by the process under control. Any of these three factors may vary, and considerable experience is required in estimating their magnitude. Every effort should be made to select the size of valve which will pass the normal operating flow at 60 to 75 per cent of maximum flow.

If the valve or other control element is oversize it will pass the normally required flow at some lower setting; e.g., if the valve has a characteristic as in Fig. 5–4 and is oversize, it may pass the normal maximum flow at 35 per cent of its maximum flow. The minimum controllable flow, however, is 2 per cent, and it represents a greater part of the normal flow. The turndown is then only 17 instead of 35. In addition, the valve is operating around the lower part of the flow characteristic. Oversizing is undesirable because in any final control element the low-flow part of the characteristic is the most irregular.

If the valve is undersize it will pass the normally required flow at a high setting, for example, at 90 per cent of its maximum setting. In the normal operation of the control system the valve is moved both above and below its normal flow position. The valve may then reach its open position without satisfying the demands of the controller if the valve is undersize.

Inner Valves

The control valve in most controlling means consists of two mechanisms, the valve body and the power unit. The valve body contains the inner valve or plug which moves over a seat or port thereby changing the area of opening through the valve. Control valves have either

sliding-stem motion for moving the plug, or rotary plug motion, in which a ported sleeve is rotated past an opening. Inner valves of the sliding-stem type are generally identified by the plug as follows:

Parabolic plug.
Developed V-port plug.
V-port plug.
Bevel plug.
Rectangular port plug.

The *parabolic plug*, illustrated in Fig. 5–7, is positioned by a sliding stem in a circular opening or seat in the valve so that the flow of fluid is

FIG. 5–7. Parabolic Plug for a Control Valve.
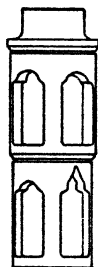
FIG. 5–8. Developed V-Port Plug for a Control Valve.
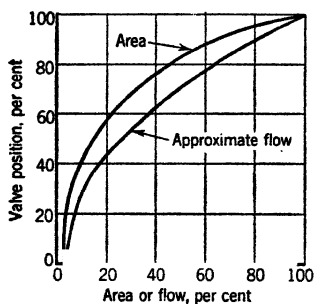
FIG. 5–9. Characteristics of Parabolic and Developed V-Port Valves.

throttled. The area characteristic of this type is semi-logarithmic or equal percentage as shown in Fig. 5–9. The plug is shaped to provide this characteristic of flow at various openings. With more than 70 per cent of the maximum pressure differential across the valve, the actual flow characteristic will be approximately as shown in Fig. 5–9.

The *developed V-port plug* has a number of ports, all of different shape as shown in Fig. 5–8. The area at any one setting is determined by the sum of all the partly open areas. The area and flow characteristics are also shown by Fig. 5–9 since the various ports are shaped to provide semi-logarithmic increases of area with position.

The *V-port plug* is similar in construction to the developed V-port plug except that all the ports are made the shape of a straight-sided V-notch as shown in Fig. 5–10. If the area of the triangular port is plotted against its height a parabolic curve is obtained such that

$$P^2 = K(F - F_0) \qquad [5\text{–}8]$$

where P = position of plug in opening.
K = constant.
F = flow through valve.
F_0 = minimum controllable flow.

When the valve position is zero the flow is F_0, which is the minimum controllable flow. The characteristic of a V-port valve approximates a
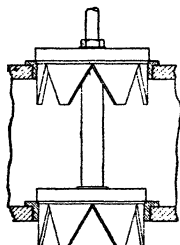
FIG. 5–10. V-Port Plug for a Control Valve.
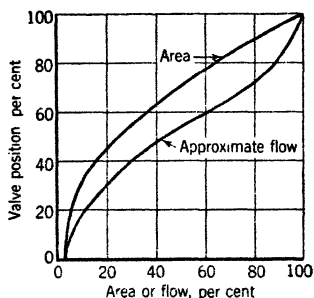
FIG. 5–11. Characteristics of a V-Port Valve.

semi-logarithmic characteristic since a parabola may be arranged to fall quite close to a semi-logarithmic curve. The area characteristic is shown in Fig. 5–11. The actual flow characteristic with more than
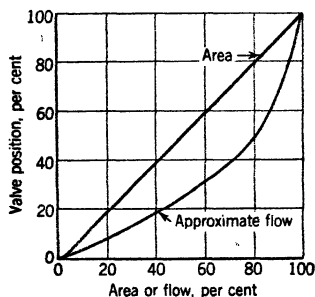
FIG. 5–12. Characteristics of a Rectangular Port and Bevel Plug Control Valves.
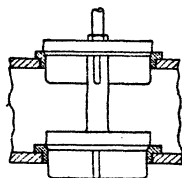
FIG. 5–13. Bevel Plug for a Control Valve.

70 per cent of the total pressure differential across the valve falls below the area characteristic.

The *rectangular port plug* operates with a short stroke or lift as distinguished from the types just discussed. The rectangular port plug is

sometimes called the quick opening or poppet type. The rectangular port plug is similar to the V-port plug except that the ports are rectangular instead of V-notch. The area characteristic in Fig. 5–12 is linear inasmuch as the area of opening increases linearly with movement of the valve stem. The flow characteristic shows decreased flow at partly open positions.

The *bevel plug* is a flat disc which is seated against a circular opening. Its construction is shown in Fig. 5–13. The area and flow characteristics are the same as for the rectangular port plug.
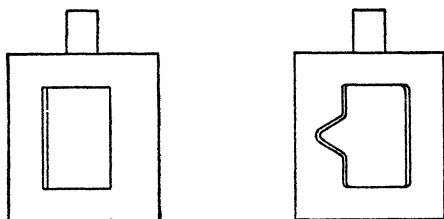
FIG. 5–14. Inner Valves for Rotary Stem Control Valves.

The inner valves of the sliding-stem type may be single seated, with only one port and seat, or double seated, with both ports opened and closed simultaneously. The double-seated valve is pressure balanced so that the forces of pressure differential on the valve stem are opposed. The resultant thrust on the inner valve is thereby reduced to a minimum regardless of the pressure differential on the plug.

The single-seated valve can be absolutely tightly shut off; a double-seated valve cannot, owing to clearances provided for the temperature expansion of materials of the valve body and plug. The clearances produce a minimum flow which cannot be controlled and determines the rangeability of the valve.

In the *rotary plug* valve the valve plug is rotated inside a cylindrical housing, as illustrated by Fig. 5–14. The rotary motion uncovers a part of the opening through the plug. The plug may have a rectangular port or a V-port, and the characteristics are the same as for the corresponding sliding-stem valve.

With rotary plug valves, and with almost any of the other types of final control elements, characterization may be obtained by means of a linkage arrangement between the power unit and valve. The purpose of using characterization is to improve the existing flow characteristic of a control element so that it is as close to linear or semi-logarithmic as possible. The need for characterization will be more apparent when the flow characteristics of the butterfly valve and louver are later considered.

Characterization is usually accomplished by arranging the angular motions of the valve operating arm and the power unit crank so that the valve is positioned in large increments near the closed position and in small increments toward the open position. Figure 5–15 shows the result of using a linkage arrangement for changing the flow characteristic.
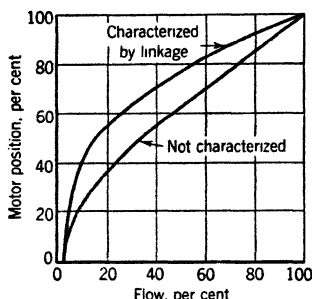
FIG. 5–15. Effect of Linkage Characterization for a Rotary Stem Control Valve.

Whenever rotary plug valves are used the linkage and mounting arrangement should be carefully selected and not haphazardly assembled since the characteristic is almost entirely dependent upon the linkage. Great care must be exerted in arranging the valve connections so that reverse characterization (large increments of motion near open position) is avoided.

The *area* characteristics just discussed represent the *flow* characteristic when the pressure differential is constant. It has been pointed out that such an ideal condition rarely exists in actual practice. Line losses and the attendant variations in pressure differentials must be considered before the actual characteristic can be determined. This can be done only when the details of the application are known.

The flow characteristics are also greatly influenced by design of the valve body, design of inner valve, and the type of power unit. If the flow characteristic should become doubtful the best means of checking the operation is to install a flowmeter and check the flow characteristic under actual operating conditions.

BUTTERFLY VALVES AND DAMPERS

The butterfly valve or damper consists of a vane rotating about its center inside a circular or rectangular casing. In many respects it resembles a stovepipe damper. The vane is operated by a shaft which projects through the casing. The butterfly valve can control the flow of liquids or gases with low pressure differentials only. High unbalanced forces are usually created with large pressure differentials, and the butterfly valve and damper are generally not suitable under these conditions. The butterfly damper is more often encountered in large sizes for the control of the flow of gases in ducts. In a 90-degree butterfly valve or damper the vane moves through 90 angular degrees to the full open position; a 60-degree butterfly valve opens only 60 degrees.

Another type of butterfly valve has its body so arranged that the vane is rotated past a V-slot in the body. This type is known as the V-port butterfly valve, and the area characteristic is nearer semi-logarithmic than the standard types.

The area characteristic of a butterfly valve or damper depends upon the construction of the vane and casing and upon the amount of rotation. The characteristic of the average 90-degree butterfly valve is approximately as shown in Fig. 5-16. As with rotary plug valves, linkage characterization may be, and generally is, used to improve this characteristic.

FIG. 5-16. Characteristics of a Butterfly Valve or Damper. (From Dickey and Coplen, *ASME Trans.*, reference 1.)

In butterfly valves and dampers as in any other valve the characteristic is dependent upon the pressure differential. Figure 5-16 illustrates how the characteristic varies at different pressure differentials and also demonstrates the fact that the actual flow characteristic under operating conditions differs from the area characteristic. At least 60 per cent of the available pressure differential should be across the valve or damper in order to maintain an effective characteristic.

The rangeability of a butterfly valve or damper is not quite as large as for a plug valve and may vary from 5 to 50. The reasons are that butterfly valves and dampers are generally employed for controlling the flow of air and other gases, sometimes at higher temperatures, sizes are much larger, and leakage is more difficult to maintain at a low value.

LOUVERS

A louver is composed of a number of adjacent rectangular vanes, similar to a venetian blind; it is illustrated in Fig. 5-17. It is used for the control of the flow of air and other gases. The three methods of arranging the louver are indicated in Fig. 5-18.

The *unirotational louver* has vanes operating always in a parallel position and all vanes rotate in the same direction. The flow characteristic is represented by curve A of Fig. 5-19, in which it will be noticed that the open position has very little effect on the flow. A movement of the vanes from 30 to 50 per cent open causes a 55 per cent change in flow so that positioning is quite critical. The critical flow characteristic of a unirotational louver is caused by the nature of flow at the open position where the fluid is merely deflected by the vanes without accomplishing

an actual reduction of flow. Figure 5–20 illustrates that the last 30 per cent of opening is not effective, especially at low velocities. The unirotational louver should be adjusted to open not more than 60 per cent, as a maximum, inasmuch as any further opening does not result in throttling of the flow.
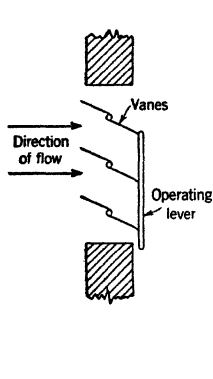
FIG. 5–17. Method of Operating a Louver.
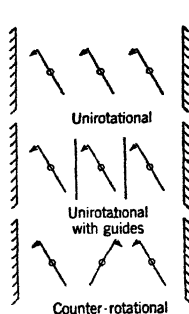
FIG. 5–18. Types of Louver Operation.
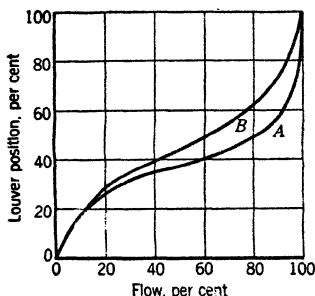
FIG. 5–19. Flow Characteristics of Louvers.
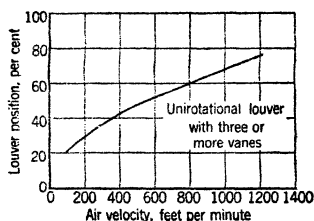
FIG. 5–20. Louver Opening to Pass 96 Per Cent Flow at Various Velocities.

The *unirotational louver with flow guides* is illustrated in Fig. 5–18. The use of flow guides improves the flow characteristic, as shown by curve *B* of Fig. 5–19, because the deflection of the fluid passing through the louver is decreased and the open position of the louver is more

effective. The flow characteristic for this type of louver is similar to the flow characteristic for the butterfly valve and damper.

The *counter-rotational louver* illustrated in Fig. 5–18 provides better control of flow because the flow through the louver can be streamlined with very little change of direction. The flow characteristic is generally about the same as for the unirotational louver with flow guides shown by curve *B* of Fig. 5–19.

The flow characteristics of louvers are affected by pressure differential, velocity, and construction more than any other type of control element. Linkage characterization together with limiting the maximum movement of the louver to 60 per cent improves the operation appreciably. The pressure differential across the louver should be at least 60 per cent of the total available differential.
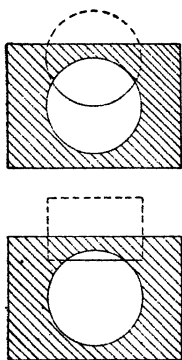
Fig. 5–21. Slide Damper Arrangements.
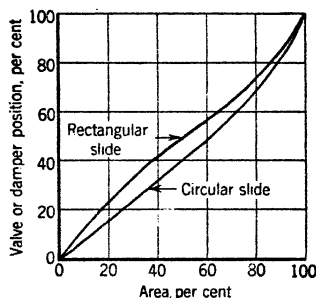
Fig. 5–22. Area Characteristics of Gate Valves and Slide Dampers.

Gate Valves and Slide Dampers

A gate valve or a slide damper has either a circular or a rectangular slide in a circular casing, as shown in Fig. 5–21. The gate valve may run in size from 2 to 24 inches, but the slide damper is usually much larger — sometimes 4 to 6 feet in diameter.

The area characteristics of these two arrangements are shown in Fig. 5–22. The rectangular slide has the better characteristic of the two and is the one more often used in large sizes. Adjustments of the linkage may sometimes be made to bring the characteristic closer to a semi-logarithmic relationship, but with the larger slide dampers this is not possible because of the heavy equipment involved.

The slide damper is generally inclined slightly from vertical in order

to obtain tight seating from the weight of the damper against the slide. The velocity of flow should be high enough to prevent dust from depositing in the valve. On the other hand, an extremely high velocity of flow should be avoided because of the turbulence which might be encountered. The flow characteristic for the slide damper should be carefully checked on the actual application, because considerable variation may be expected from differences in construction.

Power Units

A power unit is the mechanism which furnishes power for operating the valve, damper, or louver; it may be one of the following:

Electric solenoid.
Electric motor.
Pneumatic or hydraulic diaphragm.
Pneumatic or hydraulic cylinder.

The *electric solenoid* may be applied to either a sliding-stem or a rotary plug valve, and it is generally used only for two-position control. Figure 5–23 shows schematically the construction of a solenoid valve. The valve is usually of the single-seated type, and the solenoid must generate enough power to overcome the force of the spring. The speed of operation of a solenoid valve depends upon the fluid being handled by the valve but is generally less than 1 second.
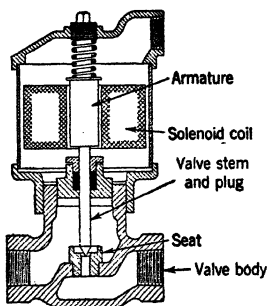
Fig. 5–23. Construction of a Typical Solenoid Valve.

The *electric two-position motor* may drive the valve open and allow a spring to close the valve, or it may drive the valve both open and closed. The spring return motor valve operates through full travel in 2 to 6 seconds. The electric motor valve driving in both directions operates through full travel in 15 to 120 seconds, depending on its gearing arrangement.

The *electric floating motor* drives the valve either open or closed with a slow speed of operation, generally greater than 120 seconds for full travel. With the floating motor the coast may cause the valve to overrun. The coast of the average floating motor is normally about 1 per cent of full travel.

The *electric proportional motor* is a reversible electric motor with a slide wire or other means added so that the valve position can be determined. The operation of this system has already been described. The speed of

operation is usually between 15 and 60 seconds but may sometimes be slower. The speed of operation should be as great as possible although the dead zone of operation is generally a cause of more controller lag than is inherent in a slow-speed motor.

The electric proportional or floating motor valve has one advantage generally not obtainable in pneumatic or hydraulic systems in that thrust caused by the unbalanced forces of a plug-type valve is not a problem because of the large gear reduction between the valve and motor. If the power fails, the valve will remain in its last position. Similarly, the valve-stem friction caused by the packing gland around the stem generally does not affect the operation of the motor valve. The coast of the motor can be compensated by damping the action of the motor as it nears the balance point.

Fig. 5–24. Construction of a Simple Pneumatic Diaphragm Valve.

The *pneumatic diaphragm valve* is shown schematically in Fig. 5–24. Air pressure is applied to one side of the diaphragm, and the force is opposed by the compression of the spring. The diaphragm motor may serve to operate either a sliding-stem or rotary plug valve. A diaphragm valve may be either direct acting, closing on an increase in pressure, or reverse acting, opening on an increase in pressure. If the air pressure fails, the force of the spring will cause the valve either to open or to close, depending upon the action of the valve.

The performance of the diaphragm valve depends upon the characteristics of the diaphragm motor as well as the valve. Figure 5–25 shows that a well-designed diaphragm motor has a linear relation between pressure and movement. The hysteresis amounts to about 2.0 per cent of the total valve travel. The effect of poor diaphragm or spring characteristics is to increase the hysteresis and change the flow characteristic of the valve.

Hysteresis is caused by stem friction in the valve as well as by poor spring and diaphragm characteristics. The stem friction may be small for valves handling water at low pressure, but it increases considerably if the control agent is corrosive or at high temperature and pressure.

With the average diaphragm valve the stem friction varies between 3 and 12 lb. The diaphragm pressure change required to overcome the stem friction would be between 0.25 and 1 per cent of full pressure change by the controller.

The thrust on the valve stem of a sliding-stem valve caused by the action of the pressure differential across a valve plug or butterfly vane
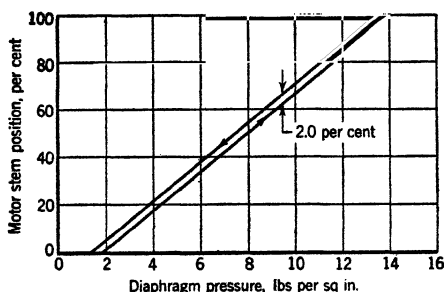
FIG. 5-25. Calibration of a Pneumatic Diaphragm Motor.

also affects the operation of the pneumatic diaphragm valve. The thrust of a single-seated valve is greater since it is not pressure balanced like a double-seated valve. With the proper design of valve body and inner valve the change in thrust can be minimized. However, if a change in thrust of 12 lb on the valve stem of a 100-sq-in. diaphragm valve occurs, the valve will change position by about 1 per cent without any counteraction.
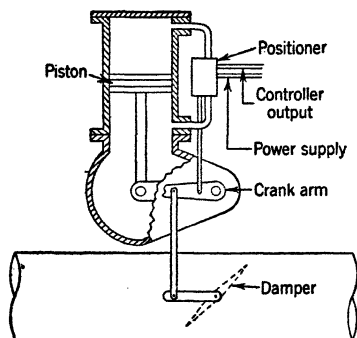
FIG. 5-26. Schematic Diagram of a Power Cylinder for Positioning a Butterfly Damper.

The *pneumatic or hydraulic cylinder*, shown schematically in Fig. 5-26, uses power on the piston in both directions for operation of the damper or valve. The hydraulic cylinder for proportional-speed floating control does not require a positioner since by supplying a hydraulic pressure in proportion to the deviation of the controlled variable the piston gradually floats in one direction or the other. The pneumatic cylinder requires a positioner for proportional control, as does the hydraulic cylinder, so as to posi-

tion the piston at any desired point and overcome the effect of unbalanced forces and friction in the final element.

When a large damper or other final element is operated by a pneumatic or hydraulic cylinder the dead weight of the element should always be counterbalanced so that the cylinder is working against only the friction of the element. The counterbalance increases the inertia of the final element, but it will result in less wear on the cylinder and the equipment that supplies the operating medium.

Positioner

A positioner is a mechanism used with a final control element for applying power whenever a change in position is required. The pneumatic-valve positioner is the most common, but a positioner is also used with the pneumatic cylinder. The electric proportional control system may be considered as inherently having a positioner in the form of the detecting unit and electric motor.

A valve positioner is applied to the pneumatic-diaphragm motor in such a way that a large air pressure is applied to the diaphragm when it is moving in one direction and is almost entirely relieved when it is moving in the other direction. When the valve has moved to the proper position the pressure is reduced to the value required to hold the position. The valve positioner, therefore, counteracts the effect of stem friction, diaphragm characteristics, and valve thrust in a pneumatic-diaphragm valve and reduces the controller lag.

The dead zone of the average valve positioner with respect to change in pressure by the controller is about 0.12 per cent of the full pressure range but varies with the amount of friction load. A stem friction of 35 lb is high, even for a valve handling acid, so that the positioner will nearly always respond to at least 0.4 per cent change in pressure from the controller. A positioner will reduce the dead zone in positioning the average control valve from 0.60 per cent down to 0.12 per cent of full pressure change — a reduction of the dead zone to one-fifth.

The hysteresis of the diaphragm valve with a positioner is reduced from more than 2.0 per cent to less than 0.20 per cent of full valve travel. Moreover, the diaphragm motor characteristics are not important with a positioner since the diaphragm and its spring serve as the power exerting means and not as the positioning means.

The thrust loads on the valve stem are compensated by the valve positioner since the positioner will force the valve to a given setting regardless of outside forces until a change in position is demanded by the controller. For example, if a 30-lb thrust load is suddenly imposed on the valve, the positioner will maintain the valve position within

about 1 per cent, whereas, if the positioner were not used, the valve would move about 3 per cent of its full travel.

The controller lag of a pneumatic control system caused by the volume of the valve top is reduced with the valve positioner. The lag between the controller and the valve positioner is small since the connecting line runs directly to a small receiving bellows in the positioner. The positioner changes the pressure on the valve faster than the controller alone can change it.

The many advantages of a pneumatic valve positioner are obvious, and, since it is not an expensive mechanism, its use should be considered whenever there is the slightest doubt concerning the ability of the valve to perform as the controller requires. If the control valve or other final element is working under corrosive or dirty conditions, or when wide proportional bands are required in the controller, the valve positioner is a necessary requisite.

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Chapter 6

PROCESS CHARACTERISTICS

Process lags are the most important characteristics affecting the application of automatic control. The physical arrangement of a process involves a well-defined law of reaction which can be expressed in terms of process lag. The reaction of the process may be very simple, or it may be quite complicated. An analysis of process reactions, requiring considerable experience and judgment, is a prerequisite to the selection of the mode of control.

A process is an operation or series of operations leading to some end result. Many conditions in the process affect the end result. One of these conditions is manipulated by the controller, and another is measured to maintain the balance of the process. For example, the flow of fuel to a heating furnace is manipulated by the controller, and the temperature of the air in the furnace is measured as indicative of the temperature of the metal being heated.

The ambient conditions must always be included in the tabulation of variables affecting the operation of the process. Very often changes in ambient conditions become large enough to affect seriously the balance within the process. These conditions are:

Atmospheric temperature.
Atmospheric pressure.
Atmospheric humidity.
Composition of atmosphere.
Wind conditions.
Solar radiation.

Process Variables

Of the many variable conditions which affect the maintenance of balance within the process, one must be selected as being indicative of or proportional to the balance. It is rarely possible to measure the actual final condition which is under control. Although the controller actually measures a temperature, a pressure, or other physical quantity, the condition being controlled may be anything for which automatic control is appropriate.

The temperature of a furnace, for example, is controlled within fixed

limits in order to produce a properly annealed metal. If the temperature went below the limits, the residual stresses of the metal might not be relieved; if it rose above the limits, the grain structure might not be satisfactory.

The basic controlled condition is the characteristics of the metal, not the temperature. Temperature is only indicative of the controlled condition. The fixed limits of temperature control are set with relation to the condition of the metal being produced.

The relationship between controlled variable and controlled condition is not always linear. In an oil-cracking furnace, for example, the rate of cracking varies approximately as the fourth power of temperature. In such applications it is always necessary to maintain very narrow limits of temperature in order to control properly the rate of cracking; the controlled condition is critical.

The selection of the controlled variable is therefore an important step in setting up the controlled system. The controlled variable should bear a definite and fixed relationship to the controlled condition of the process. If there is any change in this relationship, automatic control is incapable of correcting it.

Another of the variable conditions affecting the balance of the process must be selected for manipulation by the controller. This variable is the one in which changes in magnitude have the greatest influence upon the magnitude of the variable selected for measurement. The variable selected is generally the energy input to the process and is therefore a flow of control agent.

In a heating furnace, for example, the fuel supply is set by the controller. In a liquid-level application the flow of liquid to or from the vessel is adjusted. Sometimes, however, the control is accomplished by manipulating the flow of material, or load, to the process. For example, in controlling the amount of moisture in yarn or cloth, constant heat is supplied to drive off the moisture. The moisture content is controlled by varying the speed of the yarn through the drying machine.

The remaining variables, after the controlled variable and the control agent have been selected, may remain free. Changes in the uncontrolled variables, when they do occur, must be automatically counteracted by the control system.

Frequently the uncontrolled variables must themselves be controlled in order to make possible more exact control of the balance in the process. Sudden changes in the uncontrolled variables carry into the controlled system and cause disturbances which may keep the controller in a constantly fluctuating state. Control of the auxiliary variables should not be an afterthought but a planned part of the control system.

In the control of a fractionating column, for example, nearly every variable is under control. The temperature of the column is controlled by manipulating the flow of steam to a reboiler. In addition, the steam pressure is usually controlled as a part of the control system in order to avoid changes due to varying steam pressures. The column pressure, the rate of feed, the rate of outflow, the rate of column reflux, and the reboiler temperature are all separately controlled; very few of the process variables are left free to fluctuate.

Basic Characteristics

In any process the units of quantity, potential, and time are involved. Quantity is represented by the unit of flow supplied to the process. This may be a British thermal unit in thermal processes, or it may be a cubic foot or a pound of steam, water, or oil in other processes. Potential is the available level of energy in the process such as is represented by temperature, pressure, or head.

The combination of these units defines a characteristic known as capacity. A process absorbs and stores up energy supplied to it. This capacity is represented in simple forms by the storing of heat in a metal, by the volume of liquid in a tank, or by the storing of an electrical charge on the plates of a capacitor.

Thermal capacity is a well-known characteristic derived from specific heat. For example, the specific heat of water at standard conditions is 1.0; that is, if 1 lb of water is raised in temperature 1° F, 1 Btu will be absorbed. If 220 lb of water is heated, the thermal capacity is 220 Btu per degree. The units of thermal capacity for a specific mass are

$$C \text{ (thermal)} \propto \frac{\text{Btu}}{\text{Degree}} \qquad [6\text{-}1]$$

Similarly, we may express the capacity of a process involving pressure. For example, a pressure vessel of a certain volume may require 1 cu ft of air to raise the pressure by 1 lb per sq in. The units of pressure capacity for a specific vessel are

$$C \text{ (pressure)} \propto \frac{\text{Cubic feet}}{\text{Pound per square inch}} \qquad [6\text{-}2]$$

Pressure systems are also expressible in terms of weight, equivalent to volume, and head, equivalent to pressure. The units of pressure capacity may then be pounds per foot.

In a liquid-level process, the addition of 1 cu ft of liquid would raise

the level by 1 ft in a tank 1 ft square. The units of liquid-level capacity are

$$C \text{ (liquid level)} \propto \frac{\text{Cubic feet}}{\text{Foot}} \qquad [6\text{–}3]$$

In dealing with fluid flow, the consideration of capacity is complicated by the inclusion of time in the measured variable. Fluid flow is measured in terms of pressure inasmuch as the flowmeter is a pressure-measuring device. Therefore fluid flow may be considered a pressure type of process. If the flow of fluid in a line could be stopped without disturbing the pressure conditions, capacity would be measured in terms of the cubic feet of fluid added to the line to increase the pressure 1 lb per sq in. It is obvious that in most flow processes there is little or no capacity — only that due to compressibility of the fluid.

Another process characteristic which involves quantity, potential, and time is resistance. Thermal resistance is the reciprocal of thermal conductance. When heat is being transferred by means of conduction through a solid, a temperature drop due to the conductivity (actually the resistivity) of the solid occurs. For example, the temperature drop across an insulating wall is indicative of resistance to heat flow.

Similarly, in fluid flow there are pressure differentials over the pipe, orifice, and other restrictions to flow. The drop in potential energy is due to resistance. Electrically, this is analogous to the flow of electrical current through a wire or resistor where a voltage drop due to electrical resistance occurs at successive points. Very often process resistance is not a linear quantity since different values of resistance may be obtained at different potential levels. It is then necessary to base the process resistance on definite values of potential level.

For a heat flow through a specific body, the units of resistance are the reciprocal of thermal conductance, so that

$$R \text{ (thermal)} \propto \frac{\text{Degrees}}{\text{Btu per minute}} \qquad [6\text{–}4]$$

The units of resistance for flow or pressure where a pressure drop in pounds per square inch accompanies a flow in cubic feet per minute are

$$R \text{ (flow or pressure)} \propto \frac{\text{Pounds per square inch}}{\text{Cubic foot per minute}} \qquad [6\text{–}5]$$

For liquid level the units are similar, with the potential expressed in feet instead of pounds per square inch. Thus,

$$R \text{ (liquid level)} \propto \frac{\text{Feet}}{\text{Cubic foot per minute}} \qquad [6\text{–}6]$$

These basic characteristics of a process are summarized in the table below. Electrical units are also included for comparison. For any other process the exact meaning of the basic characteristics can be determined by a similar method of analysis.

	Dimensional Symbol	Thermal	Pressure	Liquid Level	Electrical
Quantity	W	Btu	Cubic foot	Cubic foot	Coulomb
Potential	P	Degree	Pound per square inch	Foot	Volt
Time	T	Minute	Minute	Minute	Second
Flow	$\frac{W}{T}$	$\frac{\text{Btu}}{\text{Minute}}$	$\frac{\text{Cubic feet}}{\text{Minute}}$	$\frac{\text{Cubic feet}}{\text{Minute}}$	$\frac{\text{Coulombs}}{\text{Second}} = \text{Amperes}$
Capacity	$\frac{W}{P}$	$\frac{\text{Btu}}{\text{Degree}}$	$\frac{\text{Cubic feet}}{\text{Pound per square inch}}$	$\frac{\text{Cubic feet}}{\text{Foot}}$	$\frac{\text{Coulombs}}{\text{Volt}} = \text{Farads}$
Resistance	$\frac{PT}{W}$	$\frac{\text{Degrees}}{\text{Btu per minute}}$	$\frac{\text{Pounds per square inch}}{\text{Cubic foot per minute}}$	$\frac{\text{Feet}}{\text{Cubic foot per minute}}$	$\frac{\text{Volts}}{\text{Coulomb per second}} = \text{Ohms}$

Effect of Capacity

The time effect of capacity is the determining factor in the reaction of the process. In automatic control, the continuous change brought about by the controller in maintaining the balance of the process is
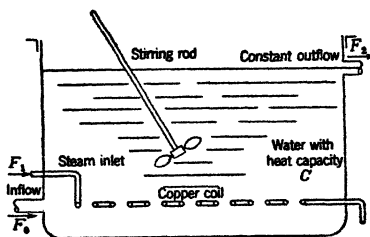
Fig. 6–1. Simple Thermal Process.
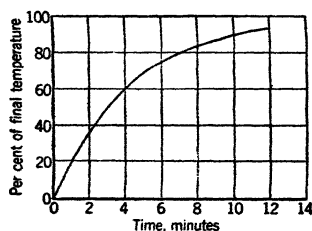
Fig. 6–2. Effect of Process Capacity.

dependent upon the relative rate at which the process can be made to react. The first step in the analysis of a process must, therefore, include an investigation of its capacity.

Let us suppose that a small amount of water is heated by means of a copper steam coil as shown in Fig. 6–1, the water being agitated. It is assumed that the thermal resistance of the thin copper steam coil is negligible. If we suddenly increase the flow of steam in the coil the water will rise in temperature. It may be shown that this rise in temperature will be in accordance with Fig. 6–2. This curve follows

Newton's basic equation, which was considered in connection with the lag of a primary measuring element.

The effect of the capacity of the water in this example is to delay the reaching of the final temperature. Theoretically the final temperature would never be attained because the curve approaches the final value asymptotically. The shape of the curve near zero time shows that a very fast change in temperature is immediately obtained.

This type of reaction is typical of a single-capacity process in which the potential of only one continuous capacity is affected by a change in energy supply. It is also typical of a single-capacity process when there is little or no resistance to flow of energy between the supply and the capacity.
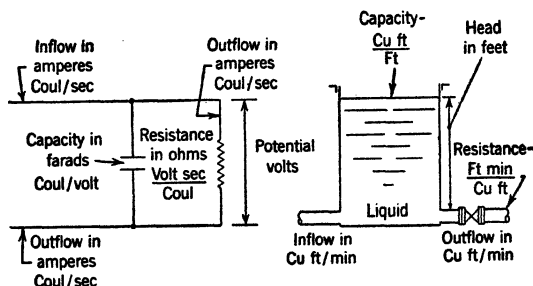
FIG. 6–3. Electrical and Hydraulic Analogies for a Single-Capacity Process.

For the reader who is familiar with electricity or hydraulics the analogous single-capacity systems are shown in Fig. 6–3. The voltage-time curve for a change in voltage over the capacitor, and the level-time curve for a change in inflow to the tank, are identical to the curve just described for the thermal process. Other simple arrangements of this nature can be set up for pressure processes where a single capacity is involved.

The reaction of a single capacity in a process depends upon both its capacity and resistance, as may be seen in the hydraulic and electrical analogies in Fig. 6–3. The lag coefficient of either of these arrangements is easily determined since it is equal to the product of resistance and capacity, or RC. This lag coefficient is the same term previously used in studying measuring lag and controller lag; is the time required to obtain 63 per cent of a complete change.

The lag coefficient or time constant of a process capacity is also related to the rate of reaction of the process.[11] In the curve of Fig. 6–2, for example, the process lag coefficient is about 4.5 minutes. The process reaction rate is the inverse of the process capacity or $1/C$. In the example above, the process reaction rate would be 0.22 per cent per

minute assuming a change in Fig. 6–3 of 100 per cent of scale. For a single capacity, maximum process reaction rate is obtained at the moment that a change in energy supply is made.

Therefore, when speaking of a " large capacity " process it is more proper to say " slow reaction " process, or process with a small reaction rate.

Transfer Lag

A process may, and usually does, have more than one capacity associated with its physical arrangement. If a flow of heat or other energy passes from one capacity through a resistance to another capacity, a lag of distinctive type is introduced; it is called transfer lag.
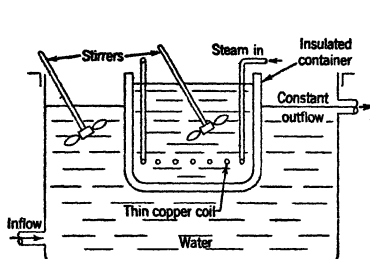
Fig. 6–4. Thermal Process with Two Significant Capacities.
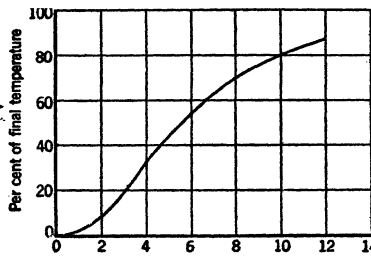
Fig. 6–5. Effect of Transfer Lag.

If a process has only one continuous capacity there can be only capacity and there cannot be any transfer lag. If a process has resistance but no capacity there cannot be either capacity or transfer lag. If a process has two capacities and one is separated by a resistance from the other, there are both capacity and transfer lag.

The heating process shown in Fig. 6–4 has two capacities separated by the resistance of the container. The difference between this process and the one of Fig. 6–1 is that heat to the water in the bottom container must pass through the water and wall of the upper container.

The temperature rise of the water in the bottom container on a change in steam supply is shown in Fig. 6–5. The marked difference in the reaction of this process is due to the addition of resistance between two capacities. The temperature does not rise instantly at its maximum rate but gradually approaches a maximum rate. Thus, the initial reaction of the process is greatly delayed by the existence of transfer lag.

The electrical and hydraulic counterparts in Fig. 6–6 also illustrate the effect of transfer lag. The transfer lag created by the resistance

between the two capacities is most clearly shown by the electrical analog. It is obvious that the capacity and resistance must be considered together since either large capacities or large resistances produce large transfer lag.
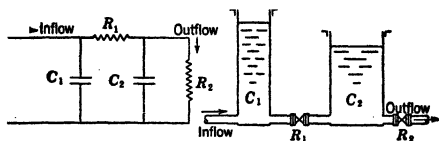
FIG. 6–6. Electrical and Hydraulic Analogies for a Two-Capacity Process.

The transfer lag of a two-capacity process is similar to the type of measuring lag generally obtained when a protecting well is used with a temperature-measuring element.

Transfer lag exists regardless of the size of the capacities as long as there are at least two. As described before, each of these capacities has associated with it a lag coefficient. Transfer lag is caused by the combination of two or more lag coefficients, so that it may be caused either by a capacity or a resistance.

Transfer lag is a maximum when the lag coefficients of all capacities, if there are more than two, are equal. Decreasing the lag coefficient of one of the capacities, regardless of whether it is nearer the supply side or nearer the load side, decreases the transfer lag. Therefore, transfer lag is small when all lag coefficients except one are very small.

MULTIPLE-CAPACITY PROCESS

Very often a process may have more than two capacities. No additional lags of any different sort are caused, but the effect of transfer lag becomes more pronounced. Generally in such processes the capacities are not arranged in a simple manner but are constructed with what can be called a *side-capacity.*

The electrically heated salt bath of Fig. 6–7 is an example of dead side-capacity. The heat from the heating element may flow directly either into the pot holding the bath or into the insulating wall. The hydraulic analogy shows this most clearly. When the electrical current to the heating elements is off, heat flow between the two capacities can occur only when one potential or temperature is appreciably lower than the other.

The baffle furnace of Fig. 6–8 is an example of live side-capacity. The flow of heat occurs either through or around a capacity in reaching the second capacity. In the hydraulic analogy the capacity of the walls is

omitted for simplicity. The heat capacity of the baffles forms the side-capacity.

The arrangements of processes with side-capacity are easily distinguished from processes where the flow of all energy is through every
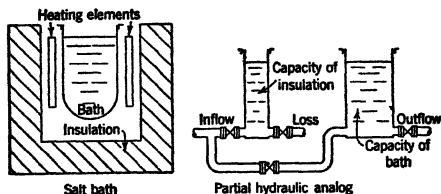
FIG. 6-7. Thermal Process and Its Partial Hydraulic Analog Showing Side-Capacity.

capacity. For example, the arrangement of the simple heat exchanger in Fig. 6-9 will have a third capacity if the heat-exchanger tubes have heavy metal walls. In this process the heat must flow through each of the capacities in series.
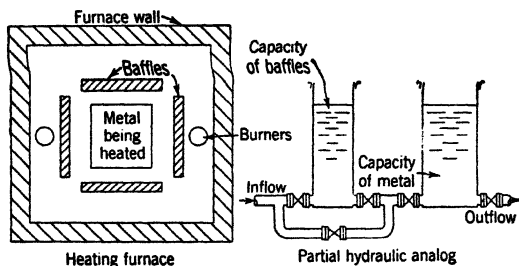
FIG. 6-8. Thermal Process and Its Partial Hydraulic Analog Showing Side-Capacity.

In some processes there may be four or five or almost any number of capacities, each separated from the others by resistance. There are innumerable arrangements, and each process must be analyzed by separating its individual capacities and resistances. A curve showing the reaction of a multiple-capacity process is of the same general type as those illustrating transfer lag.

In most thermal processes, capacities and resistances are not separable but are homogeneously distributed in a single mass. For instance, when a metal rod is heated at one end, heat flows through the rod at a rate dependent upon both the thermal capacity and the thermal conductivity of the rod. Such an example may be analyzed by considering

the rod to consist of a finite number of small capacities and resistances. Unless unusual accuracy is required, a division into approximately five or six sections is sufficient.

Other problems which must be considered with the arrangement of the capacity and resistance of a process are the non-linearities introduced by
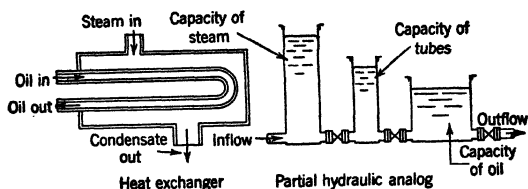
Fig. 6–9. Thermal Process and Its Partial Hydraulic Analog Showing Three Capacities in Series.

the variations in specific heat and thermal conductivity. For example, the thermal conductivity of a metal varies considerably between low and high temperatures. Non-linearity also exists in fluid flow, which is a one-half-power relationship, and radiation, which is a fourth-power relationship.

Dead Time

In certain processes, particularly continuous ones, where it is necessary to transfer heat or other energy by means of a fluid flowing through some distance at a finite velocity, a type of delay called dead time is often encountered.
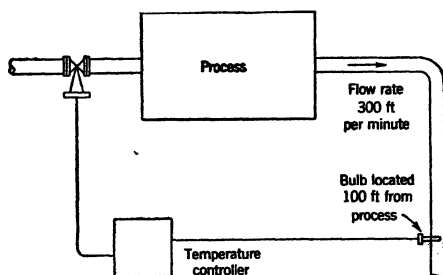
Fig. 6–10. Dead Time Caused by the Method of Installing a Temperature Controller.

Dead time is often created by installing the control valve or measuring element at a distance from the process. For example, if the fluid is flowing from the process of Fig. 6–10 at a rate of 300 ft per min, then a delay

of 0.33 minute will elapse before any change in the process is detected by the temperature-measuring element. The dead time is therefore 0.33 minute.

Dead time between the process and the measuring element is more serious than a lag located at almost any other point in the controlled system. The measurement of the balance of the process may be appreciably in error if a time delay exists between changes and their indication. The dynamic error thus caused creates considerable difference between the indicated magnitude of the variable at the controller and the actual magnitude of variable at the process.

Dead time may exist inside the process as well as outside. The continuous flow of materials in the process over an appreciable distance may cause dead time. For example, if the flow of heated air or water in a convection heating system is slow, a time delay will result, depending upon the distance to be traveled. Another application in which dead time may often be found is the concurrent heat exchanger. In the last pass the temperature levels are nearly constant. Consequently, the time required to pass this section constitutes a delay during which automatic control has no effect.

FIG. 6-11. Shift of Process Reaction Curve Caused by Dead Time.

In processes having chemical reactions it is often necessary to wait for the complete reaction to take place. The reaction period constitutes a dead time since very often an appreciable time must elapse before the reaction can be detected. Processes involving acidity or *p*H often require time for mixing of reagents. The mixture of solid reagents to form a solution is another process requiring time for completion of reaction.

Dead time causes no change in the process reaction characteristic but merely delays or shifts the reaction to a later time. Figure 6-11 illustrates a single-capacity process with and without dead time. With dead time in the process a change is not indicated until the required length of time has passed. This is the most serious lag for automatic control, since whatever happens during the dead-time period is inevitable and the control action is correspondingly delayed.

SELF-REGULATION

This characteristic, inherent in the arrangement of the process, is the tendency of the process to regulate its own potential level. Consider the

two storage tanks of Fig. 6–12. In the tank on the left, a rise in level in the tank creates a greater pressure head over the outflow valve. As the pressure head over the outflow valve increases, the flow through the valve increases. As the pressure head decreases, the flow through the valve decreases. The changes in level are thus slightly held in check by the inherent action of the pressure head over the valve.

In the tank on the right, the outflow from the process is constant. A rise in level in the process has no effect on outflow, and there is no
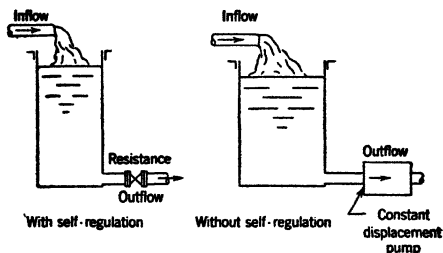
Fig. 6–12. Self-Regulation in a Liquid-Level Process.

tendency toward self-regulation. The processes shown in Fig. 6–12 do not have inflow self-regulation, and only one has outflow self-regulation. Thus self-regulation may exist at both the input and output ends of the process. Inflow self-regulation could be created by moving the inflow pipe to the bottom of the tank opposite the outflow pipe.

Most two-capacity processes exhibit a small amount of self-regulation. For example, when the temperature of the medium being cooled in a heat exchanger drops, the temperature differential between this medium and the cooling water is lower. This causes a decrease in rate of heat transfer due to the lower differential. The tendency for the temperature to drop is decreased. An opposite reaction is obtained for a temperature increase. Self-regulation of this type is caused by the resistance located between the two capacities.

Process self-regulation can also be caused by other factors. Energy losses from the process may vary directly as the magnitude of the potential. For example, the radiation, conduction, and convection losses of heat through the walls of a furnace increase as the temperature increases.

Self-regulation may be an important factor influencing the arrangement of the control system. Its magnitude is generally not sufficient to make the process completely self-controlling because a considerable deviation of controlled variable is required before any degree of counteraction is obtained. Generally enough self-regulation is present for its

action to be relied upon as a supplementary aid in establishing stability in the controlled system.

Analysis of Processes

Let us analyze the reaction characteristics of a typical process in the light of the capacity and resistance, and correlate the overall effect. For the sake of simplicity, consider the single-capacity process of Fig. 6–13 which was previously used to illustrate the effect of capacity. This process may be analyzed by considering the relationships between the inflowing and the outflowing heat.[7]

Fig. 6–13. Energy Flow Diagram for a Single-Capacity Process.

If an instantaneous change is made in the inflow of heat the temperature θ of the liquid will rise or fall at some definite rate of change. In equation form,

$$C\frac{d\theta}{dt} = F_1 - F_2 \qquad [6\text{–}7]$$

where C = capacity of liquid in container, Btu per degree.
θ = temperature of liquid, degrees Fahrenheit.
F_1 = heat supplied by steam, Btu per minute.
F_2 = heat carried away in liquid, Btu per minute.

However, the outflowing heat F_2 is affected by the temperature of the liquid. Since the temperature θ and flow F_2 are known, the outflow resistance may be found by

$$R = \frac{\theta}{F_2} \qquad [6\text{–}8]$$

or

$$F_2 = \frac{\theta}{R} \qquad [6\text{–}9]$$

where R = outflow resistance in degrees per Btu per minute. This concept of a thermal resistance where there may not be an actual resistance is a necessary one in process analysis. It arises by virtue of flow and potential at one point in an energy-transfer system, and the resistance is simply the ratio of potential to flow. The temperature which the

process will finally reach is expressible in terms of the resistance R as follows:

$$\theta_F = R(F_0 + F_1) \qquad [6\text{–}10]$$

where θ_F is the final temperature of the liquid in degrees and F_0 is the heat carried by the inflowing liquid. We may now substitute equation 6–9 in 6–7, and

$$C\frac{d\theta}{dt} = F_1 - \frac{\theta}{R} \qquad [6\text{–}11]$$

Simplifying,

$$RC\frac{d\theta}{dt} + \theta = RF_1 \qquad [6\text{–}12]$$

This equation shows that both the temperature and the rate of change of temperature depend upon the supply of heat to the process. However, equation 6–10 may be substituted in equation 6–12. Assuming that the inflowing liquid contributed no heat to the process, $F_0 = 0$, then,

$$RC\frac{d\theta}{dt} + (\theta - \theta_F) = 0 \qquad [6\text{–}13]$$

The constant θ_F may be added to the differential so that

$$RC\frac{d}{dt}(\theta - \theta_F) + (\theta - \theta_F) = 0 \qquad [6\text{–}14]$$

This equation completely describes the action of the process on a sudden change in supply of heat. The solution is easily determined since it is a linear differential equation of the first order. A particular solution is

$$\theta - \theta_F = (\theta_0 - \theta_F)e^{-\frac{t}{RC}} \qquad [6\text{–}15]$$

where $(\theta_0 - \theta_F)$ is the difference between original and final temperatures of the liquid.

It should be noted from equation 6–15 that the only factors affecting the rate at which the process comes to balance are the resistance and capacity of the process. Since these factors appear as R times C, their relative magnitudes must be considered together. The magnitude of the heat inflow, however, has no effect on the time required for the process to attain a balance but only determines the final balance point.

The reaction curve for the process can now be drawn by using an actual example. If there is 50 lb of water in the container, the capacity

is determined from the specific heat, which is 1.0, and the weight of water,

$$C = 50 \text{ Btu per degree}$$

If the rate of outflow is 10 lb per min, then the heat outflow is $(100 - 32) \times 10 = 680$ Btu per min when the original temperature θ is 100° F. Therefore, by equation 6–8,

$$R = \frac{100}{680} = 0.15° \text{ per Btu per min}$$
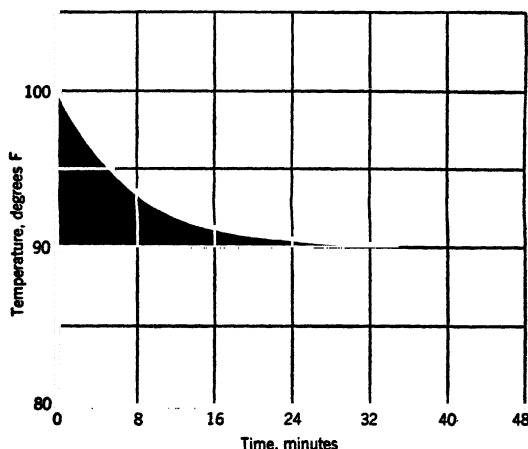
FIG. 6–14. Calculated Reaction Curve for the Single-Capacity Process of Fig. 6–13.

If the inflow of heat F_1 in the steam is suddenly changed from 680 to 600 Btu per min, the final temperature is found by equation 6–10 when the inflow water at F_0 is at 32° F so that $F_0 = 0$,

$$\theta_F = 0.15 \times 600 = 90° \text{ F}$$

The equation for the reaction of the process then becomes

$$\theta - 90 = (100 - 90)e^{-\frac{t}{7.5}} \qquad [6\text{–}16]$$

and

$$\theta = 90 + 10e^{-\frac{t}{7.5}} \qquad [6\text{–}17]$$

This equation is plotted in Fig. 6–14. The reaction is typical of a single-capacity process where the effect of the capacity is to retard the temperature change.

The conclusions which may be drawn from the analysis are:

1. The time for the process to reach balance depends only upon the capacity, initial temperature, and heat outflow.

2. Rate of temperature change depends on capacity and difference between heat inflow and heat outflow.

3. The final temperature depends upon the heat inflow assuming that a resistance to heat outflow exists.
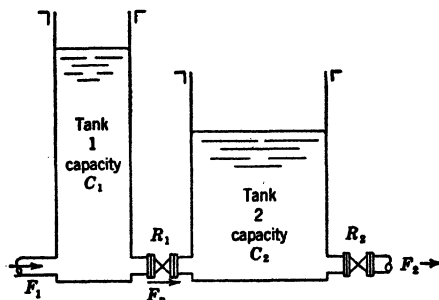
FIG. 6–15. Energy Flow Diagram for a Two-Capacity Process.

For a second example, let us investigate the two-capacity process of Fig. 6–15. The outflow of each tank occurs through valves which provide a resistance to flow. It will be necessary to assume that the flow through the resistances is linear with pressure differential in order to avoid variable coefficients in the equations. This assumption does not cause serious error when changes in the process are small and the levels in the tanks do not vary over a wide range.

The level in tank 1 assumes an instantaneous rate of change when the control valve is suddenly opened and a change of inflow made. Thus,

$$C_1 \frac{d\theta_1}{dt} = F_1 - F_R \qquad [6\text{–}18]$$

and tank 2 acts similarly at a sudden change of inflow.

$$C_2 \frac{d\theta_2}{dt} = F_R - F_2 \qquad [6\text{–}19]$$

where C_1 = capacity of tank 1, in cubic feet per foot.
C_2 = capacity of tank 2, in cubic feet per foot.
θ_1 = level in tank 1, in feet.
θ_2 = level in tank 2, in feet.
F_1 = inflow, in cubic feet per minute.
F_2 = outflow, in cubic feet per minute.
F_R = flow between tanks, in cubic feet per minute.

Owing to the resistances R_1 and R_2, the levels in each tank can be expressed by

$$\theta_2 = R_2F_2 \qquad [6\text{–}20]$$

and

$$\theta_1 = R_2F_2 + R_1F_R \qquad [6\text{–}21]$$

or, in the differential form,

$$\frac{d\theta_2}{dt} = R_2\frac{dF_2}{dt} \qquad [6\text{–}22]$$

and

$$\frac{d\theta_1}{dt} = R_2\frac{dF_2}{dt} + R_1\frac{dF_R}{dt} \qquad [6\text{–}23]$$

The rates of change of the levels, equations 6–18 and 6–23, may be equated

$$\frac{F_1 - F_R}{C_1} = R_2\frac{dF_2}{dt} + R_1\frac{dF_R}{dt} \qquad [6\text{–}24]$$

and, similarly, equations 6–19 and 6–22

$$\frac{F_R - F_2}{C_2} = R_2\frac{dF_2}{dt} \qquad [6\text{–}25]$$

Solving equation 6–25 for F_R,

$$F_R = C_2R_2\frac{dF_2}{dt} + F_2 \qquad [6\text{–}26]$$

Differentiating equation 6–26,

$$\frac{dF_R}{dt} = C_2R_2\frac{d^2F_2}{dt^2} + \frac{dF_2}{dt} \qquad [6\text{–}27]$$

Equations 6–26 and 6–27 may now be substituted in equation 6–24, thereby eliminating the flow between capacities (F_R) from consideration.

$$\frac{F_1}{C_1} - \frac{C_2}{C_1}R_2\frac{dF_2}{dt} - \frac{F_2}{C_1} = R_2\frac{dF_2}{dt} + R_1C_2R_2\frac{d^2F_2}{dt^2} + R_1\frac{dF_2}{dt} \qquad [6\text{–}28]$$

Rearranging the above equation and collecting terms,

$$(R_1C_1R_2C_2)\frac{d^2F_2}{dt^2} + (R_1C_1 + R_2C_1 + R_2C_2)\frac{dF_2}{dt} + (F_2 - F_1) = 0 \qquad [6\text{–}29]$$

After multiplying through by R_2, equations 6–20 and 6–22 may be sub-

stituted. Then

$$(R_1C_1R_2C_2)\frac{d^2\theta_2}{dt^2} + (R_1C_1 + R_2C_1 + R_2C_2)\frac{d\theta_2}{dt} + \theta_2 - R_2F_1 = 0 \qquad [6\text{-}30]$$

This linear differential equation of the second order describes the action of the process.

As in the single-capacity example, the final level to which the second tank will balance is

$$\theta_F = R_2F_1 \qquad [6\text{-}31]$$

Substituting in equation 6–30,

$$(R_1C_1R_2C_2)\frac{d^2}{dt^2}(\theta_2 - \theta_F) + (R_1C_1 + R_2C_1 + R_2C_2)\frac{d}{dt}(\theta_2 - \theta_F) + (\theta_2 - \theta_F) = 0 \qquad [6\text{-}32]$$

Actual quantitative values may now be substituted and the reaction for a particular process obtained. Suppose that:

1. Tank 1 has 40 sq ft area, $C_1 = 40$ cu ft per ft.
2. Tank 2 has 40 sq ft area, $C_2 = 40$ cu ft per ft.
3. Flow F_2 is 700 cu ft per min for a 140-ft head differential. Thus,

$$R_2 = \frac{140}{700} = 0.20\,\frac{\text{Ft min}}{\text{Cu ft}}$$

4. Flow F_R is 700 cu ft per min for a 70-ft head differential. Thus,

$$R_1 = \frac{70}{700} = 0.10\,\frac{\text{Ft min}}{\text{Cu ft}}$$

If the process was previously at balance and the inflow was suddenly decreased from 700 to 600 cu ft per min, the constants in the equation may be determined. Equation 6–32 becomes

$$32\frac{d^2}{dt^2}(\theta_2 - \theta_F) + 20\frac{d}{dt}(\theta_2 - \theta_F) + (\theta_2 - \theta_F) = 0 \qquad [6\text{-}33]$$

A particular solution for this second-order differential equation is

$$\theta_2 = 120 - 2.13e^{-0.571t} + 22.13e^{-0.055t} \qquad [6\text{-}34]$$

This equation is plotted in Fig. 6–16. As in the previous example, the coefficients of the equation describing the system are composed of the products of capacity and resistance. Therefore, the transfer lag depends not only upon the size of capacities but also upon the size of the resistances. The transfer lag is clearly shown by the reaction curve.

The final balance point is determined by the magnitude of the inflow and the outflow resistance. The magnitude of inflow has no effect on the rate at which the process approaches balance.
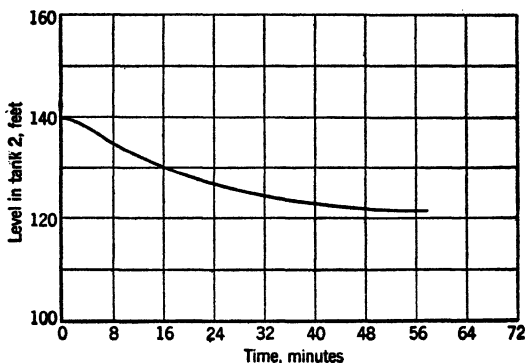
FIG. 6–16. Calculated Reaction Curve for the Two-Capacity Process of Fig. 6–15.

This method of analyzing processes is valuable where a specific application is under investigation. It is the only method in which all the important factors involved in automatic control can be included.

REACTION CURVE

One method of determining the characteristics of a process is to observe the dynamic reaction of the process to a sudden change at the supply side. In considering the measuring element, controller, and process characteristics in this and preceding chapters, this method has been employed to obtain the reaction or response characteristics.

If the valve at the supply side of the process is opened a slight amount, say 5 per cent of its total motion, then the record at the controller will change. During this time no automatic control is applied to the process, the control valve being manipulated by hand and the controller serving as a recorder. The rise or drop of the measured variable is allowed to continue until a stabilized value has been reached. The resultant curve obtained by the recording controller may be plotted on a graph showing the magnitude of the variable against time. The curve thus obtained is the reaction curve for the process.

It is important to note that this reaction curve will contain the lag of the process as well as the lag of the measuring means of the controller. If the valve is set by remote means the reaction curve will also include the controller lag. If the measuring and controller lags are small in

comparison to the process lag, the reaction curve is generally indicative of the process reaction. If the process lag is relatively small, the reaction curve illustrates the measuring and controller lags.

Dead time exists if any appreciable length of time elapses before the slightest change begins to occur in the measured variable. Figure 6–17 presents an example. A dead time of 1 to 2 minutes is worthy of note, and 5 minutes or more will make control of the process difficult. Dead time may be caused by the periodicity of a measuring or controlling means of a controller, by a dead zone in the measuring or control means, or by a flow of material in the process.
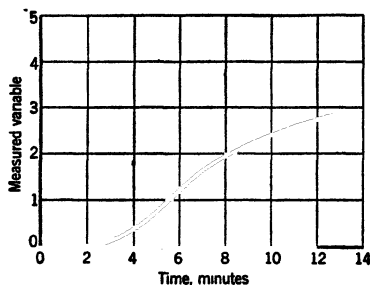
FIG. 6–17. Typical Reaction Curve Showing Dead Time and Transfer Lag.

Dead time is not often found in flow, pressure, or liquid-level processes except where the final control element is installed at considerable distance from the process. It is more likely to occur in thermal processes of the continuous type where convection methods of heat transfer are used. Radiant heat transfer nearly always eliminates any dead time unless it is inherent in the design of the process.

After the change has once begun there are almost infinite characteristics which the curve can assume. In single-capacity processes a sharp rise is obtained immediately and the variable gradually approaches a constant value. A larger capacity causes a slower increase and a greater time is required to approach a constant value, as shown by Fig. 6–18. This type of curve is generally not obtained from a thermal process but is more characteristic of a liquid-level or pressure process having only one significant capacity.
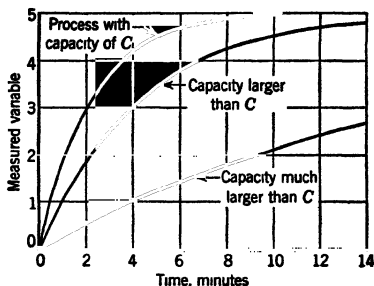
FIG. 6–18. Typical Reaction Curve Showing Effect of Process Capacity.

The sharp rise at the beginning of the change indicates that very little transfer lag is present. The measuring and controller lags must therefore be small. A pressure process generally exhibits this reaction. The maximum rate of change is high, and the time scale of Fig. 6–18 is graduated in tenths of minutes. Liquid-level processes having single

capacity, and where measuring lag is small, may have a similar reaction. The capacity may be almost any value, depending upon the size of the tank or vessel. In order to obtain a similar reaction with thermal processes, the process must be especially designed for fast reaction and the measuring means must be especially responsive. For example, high-speed salt baths with radiation measurement approach this characteristic.

A flow process usually has very little capacity. Appreciable capacity exists only in processes where the outflow from a vessel is controlled by controlling the inflow. A flow process has resistance due to the flow through the valve and orifice. When no capacity is associated with this resistance there can be no retardation due to capacity. Therefore, the change in flow rate follows the positioning of the control valve quite closely. The measuring and controller lags generally make up the total lag in a flow-control process. A reaction curve similar to Fig. 6–18 is obtained when the measuring lag is the principal one. The reaction curve has a somewhat lower rate of change for flow than for pressure when the mercury manometer flowmeter is used.

FIG. 6–19. Typical Reaction Curve Showing Effect of Transfer Lag.

As practically all thermal processes involve more than one capacity, the characteristics of a transfer lag may be noted from the reaction curve. The changing acceleration or point of inflection of the curves of Fig. 6–19 indicates multiple capacities. Notice that the curves are concave upward at the beginning of the change.

A liquid-level process having more than one capacity will show this type of reaction. The maximum rate of change depends upon the size of capacities, resistances, and magnitude of disturbance, which may vary greatly for different liquid-level processes. This same reaction can also be obtained from a single-capacity liquid-level process when the measuring means of the controller has an appreciable lag. Flow and pressure processes do not usually show this type of reaction.

The distributed capacity and resistance, and the multiple-capacity arrangements, of thermal processes cause the reaction curve to indicate transfer lag. The capacity of thermal processes is generally large, and the time scale of Fig. 6-19 may be multiplied by 10 for some applications.

The measuring and controller lags affect the reaction curve even when the capacity of the process is large, however.

When a thermometer or thermocouple in a heavy well is used to measure the temperature of the heated air or other gas the measuring means has transfer lag and the reaction curve is similar to the one of Fig. 6–19. Here the process lag is almost entirely composed of the measuring lag.
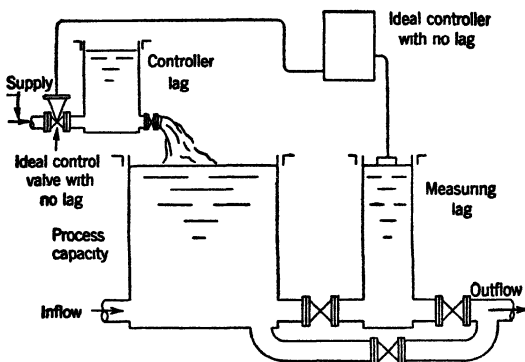
FIG. 6–20. Three-Capacity Controlled System Consisting of a Controller Lag, a Single-Capacity Process, a Measuring Lag, and an Ideal Controller.

Shown in Fig. 6–20 are the hydraulic analogies for a single-capacity process with a measuring lag and a controller lag. Note that the total number of capacities is three. The reaction curve for such a controlled system indicates transfer lag even though there is no transfer lag in the process itself.

The following table is helpful in visualizing a comparison of the majority of processes. Not every process will follow this table, but on the average it shows what can be expected. The measuring and controller lags are not included in this summary.

	TEMPERATURE	FLOW	PRESSURE	LIQUID LEVEL
	Usually 2 or more capacities	*Usually 0 capacity*	*Usually 1 capacity*	*Usually 1 capacity*
Capacity	Medium or large	Negligible	Small or medium	Small or medium
Transfer lag	Medium or large	Small	Very small	Small or medium
Dead time	Small or large	Small	Small	Small

In studying the reaction curve of a process, considerable information of value concerning the process characteristics may be obtained. In a general way, the analysis of the reaction curve indicates the complexity of the control problem. Two characteristics which are very important to the analysis of the controlled system are shown directly by the reaction curve, namely, the dead time and the transfer lag.

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Chapter 7

THEORY OF AUTOMATIC CONTROL

The problems of automatic control originate in the necessity for counteracting lags in various parts of the controlled system. The fundamental rules governing the action of the controlled system can be determined from the nature of the process lag, measuring lag, and controller lag. The mode of control, however, has the most decisive influence upon the results of the controller action.

One mode is generally more useful than another in producing the desired kind of control. The capabilities and the limitations of each mode must be assessed in order to apply automatic control to a process. The choice of a more complicated mode of control than is necessary is inconsistent with keeping the entire system as simple as possible.

The dynamic action of the controlled system, often described as a series of events, may be either stable or unstable. Stable action results when the various forces in the controlled system oppose each other with the proper timing and magnitude. Unstable action is caused by the unbalance of parts of the system and must be avoided at all costs.

Two-Position Control

Because of its simplicity, two-position control by either pneumatic, electric, or direct mechanical means is very popular. Its control action is essentially cyclic, although under almost ideal conditions the magnitude of variation in controlled variable may be made extremely small. Two-position control will be considered as having a differential, since in only a few industrial controllers is it sufficiently small to be called zero.

An *ideal case* will be considered first by supposing that a two-position controller is applied to a single-capacity process without self-regulation. The controller is assumed to have no measuring lag and no controller lag. This problem is similar to the control of liquid level in an open tank by admitting water at the upper end of the tank.

If the controller has a differential, the liquid level will rise at a constant rate until it reaches the upper edge of the differential. At this moment the valve admitting water is instantly closed. The level immediately reverses and drops at a constant rate until the lower edge

of the differential is reached, where the level is again reversed. As shown by curve A of Fig. 7–1, the controlled variable cycles between the limits of the differential.

With a process capacity as shown in curve AA of Fig. 7–1, the period of cycling is comparatively small. If the process capacity is doubled so that the rate of change of level is one-half its former value, as in curve BB, the period of cycling is doubled, as shown in curve B. The double amplitude of cycling remains unchanged since it is numerically equal to the width of differential.

If the differential is reduced to one-half of that in curve B, the period and the amplitude are reduced to one-half, as shown in curve C.
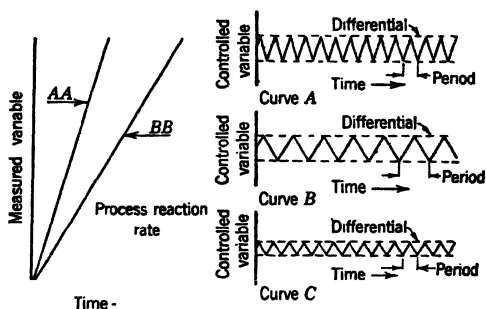
FIG. 7–1. Effect of Process Reaction Rate in Two-Position Control.

It is assumed in Fig. 7–1 that the process reaction rate is the same for both increasing and decreasing magnitudes of controlled variable. In many processes these reaction rates may be different and the cycle becomes uneven. This does not alter the action of the two-position differential controller but affects only the cycle of controlled variable in that increases and decreases in controlled variable occur at different rates.

Thus, ideally, the double amplitude of cycling is always equal to the width of controller differential. The period of cycling is inversely proportional to the reaction rate of the process and directly proportional to the width of controller differential.

Now suppose that the process or the control system possesses dead time as shown by the process reaction curve BB of Fig. 7–2 as compared to curve AA. With two-position control, the cycle of controlled variable is wider than the differential of the controller, as shown by curve B. Obviously, since dead time is present, the controlled variable continues to change until the dead-time period has elapsed and corrective action becomes effective.

If the controlled system has an appreciable dead time, the amplitude of cycling is directly proportional to the reaction rate of the process since a greater reaction rate will allow a greater " overshoot " before the controlled variable can be reversed. The amplitude of cycling is also directly proportional to the length of dead time and to the width of controller differential.
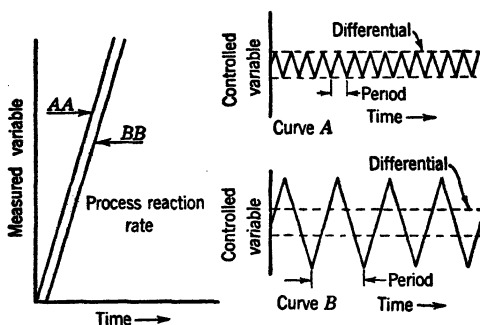
FIG. 7-2. Effect of Dead Time in Two-Position Control.

The period of cycling, when dead time is present, is inversely proportional to the reaction rate of the process, directly proportional to the width of controller differential, and directly proportional to the length of dead time.
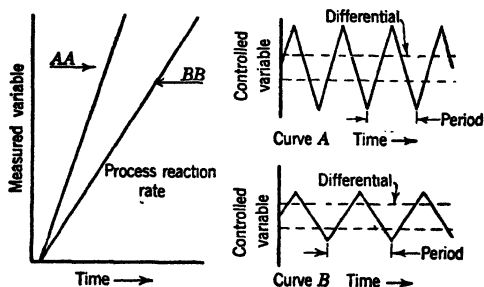
FIG. 7-3. Effect of Process Reaction Rate in Two-Position Control When Dead Time is Present.

When dead time is present in the controlled system, process capacity and the corresponding process reaction rate greatly influence the amplitude of cycling. In Fig. 7-3, two controlled systems having the same dead time but different reaction rates are shown. When two-

position differential control is used with the process having the larger reaction rate (curve *AA*), the double amplitude of cycling is 1.7 units. If the process reaction rate is reduced to one-half (curve *BB*), the double amplitude of cycling is reduced to 1 unit.

The reaction rate of the process is dependent not only upon the process capacity but also upon the amount of energy supplied by the final control element. For example, if the flow of fuel supplied by a control valve is reduced by adjusting lower and upper limits of valve
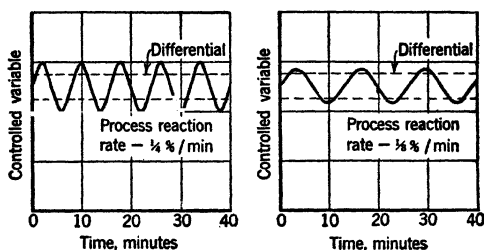
FIG. 7–4. Effect of Process Reaction Rate in Two-Position Control When Transfer Lag is Present.

opening, the process reaction rate is correspondingly reduced. By reducing the operating range of the final control element, the process reaction rate is reduced and the amplitude of cycling with two-position control is correspondingly reduced, much the same as shown in Fig. 7–3.

The *practical case* differs from the ideal because of the transfer lag introduced by the measuring and controller lags. In addition the process itself is also likely to possess transfer lag, especially since two-position control is widely employed for thermal processes.

Under these circumstances, the process reaction rate has considerable influence on the amplitude and period of cycling, as shown by Fig. 7–4. Note that the process having a slower reaction rate holds considerably closer to the differential width than the process having a faster reaction rate. The period of cycling is appreciably increased by a smaller reaction rate of the process. The transfer lag in the controlled system " rounds off " the peaks of the controlled variable cycle which, in many applications, may appear similar to a sine wave.

Varying load conditions in the process produce an unsymmetrical appearance of the cycle when two-position differential control is applied to controlled systems with transfer lag. In a thermal process, for example, a greater demand for heat must be accompanied by greater

"on" time of the final element and a shorter "off" time. Figure 7–5 shows that the average magnitude of the controlled variable must be lower when the demand is high in order to provide a higher rate of heat supply. An overshoot of the controlled variable is obtained at the time of change in demand because the average rate of heat supply is inadequate for the new demand.
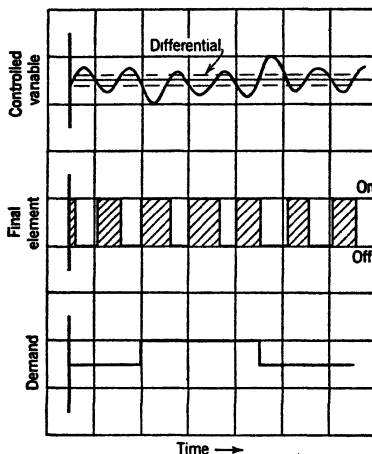
FIG. 7–5. Effect of Load Change in Two-Position Control.

Very often the process possesses more than one capacity which, in addition to measuring and controller lags, causes a sufficient degree of transfer lag that the control action becomes unstable. Then the amplitude of cycling with two-position control becomes increasingly greater until some large finite amplitude is obtained.

From the foregoing study we may draw some general conclusions regarding two-position control:

The period of cycling:

1. Increases with slower process reaction rate.
2. Increases with greater transfer lag.
3. Increases with greater dead time.
4. Increases with greater width of controller differential.

The amplitude of cycling:

1. Increases with faster process reaction rate.
2. Increases with greater transfer lag.
3. Increases with greater dead time.
4. Increases with greater width of controller differential.

Thus it is obvious that certain limitations exist in the application of two-position control. Assuming that the measuring and controller lags are not negligible, as they seldom are in temperature control, the process must possess certain so-called desirable characteristics.

The most important requirement for obtaining a small amplitude of cycling of the controlled variable is that transfer lag and dead time shall be as small as possible. With processes having large reaction rate, the dead time must be nearly zero. A slower process reaction rate allows a slight dead time without exceeding close limits of cycling of the controlled variable.

Transfer lag, whether caused by multiple capacities in the process or by measuring or controller lags, also results in greater amplitude of cycling. A large measuring lag, however, is likely to produce a serious dynamic error wherein the recorded variable does not adequately represent the actual magnitude of the variable.

A slow process reaction rate allows closer limits of two-position control. If the period of cycling should become too long owing to an excessively slow process reaction rate, then the period of cycling may be so large that an undue length of time elapses before recovery can be made from process load changes or control point shifts. A period of 30 minutes, for example, may result in a 1- or 1.5-hour time interval for stabilization of the controlled variable.

Summarizing, two-position control is most satisfactory when:

1. Transfer lag and dead time are negligible.
2. Process reaction rate is slow.
3. Measuring and controller lags are small.
4. Load changes are not large or frequent.

Immediate reaction of process and controller is of paramount importance in two-position control. As long as this action is obtainable the actual process reaction rate and process load changes may be either large or small.

Single-Speed Floating Control

Like two-position control, single-speed floating control is essentially cyclic in nature. Under certain conditions, however, stable operation may be obtained such that the controlled variable remains in the neutral near the control point.

An *ideal case* in single-speed floating control exists when the controlled system possesses no transfer lag or dead time, and the controller has a neutral zone. Under these circumstances, a movement of the controlled variable to the edge of the neutral causes a valve motion

and a quick reversal of the change in controlled variable. In this respect it is similar to two-position control, and the amplitude of cycling is equal to the width of neutral.

In order to stabilize the controlled variable within the neutral and avoid continuous cycling, the process should possess self-regulation. This is true, in general, for all types of floating control. The self-regulation of processes provides a relation between magnitude of controlled variable and value of flow of control agent. Floating control
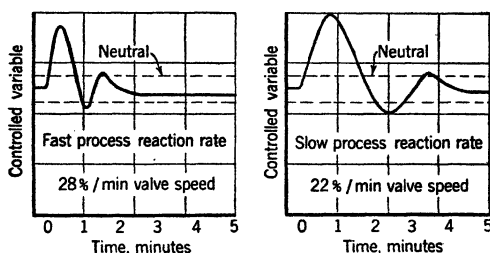
FIG. 7–6. Effect of Process Reaction Rate in Single-Speed Floating Control.

may take advantage of process self-regulation by providing the required flow of control agent to maintain the controlled variable in the neutral. If there is no self-regulation, the final element must be constantly changing the flow of control agent because of the lack of relation between value of flow of control agent and magnitude of controlled variable.

The process reaction rate in floating control must be as large as possible. This is generally brought about by a small process capacity. Figure 7–6 illustrates the longer period of cycling caused by a slower process reaction rate. Notice also that the floating speed must be appreciably decreased when the reaction rate is slow. The disadvantage of a slow process reaction rate is that the period of cycling may become very long. Then control of the variable is not so effective, and correcting for changing loads is more difficult.

In the *practical case* transfer lag is generally present either in the process or in the form of measuring lag. The process lag, caused by transfer lag and dead time, results in excessive overshooting and wide cycling. A neutral is generally useless under these circumstances since the transfer lag and dead time do not allow a stable condition to be obtained. If these lags are large the controlled variable may cycle with constantly increasing amplitude. In this respect single-speed floating control is similar to two-position control.

The floating speed must be neither too fast nor too slow if the most effective control is to be obtained. Figure 7-7 shows the effect of various floating speeds when a load change occurs in single-speed floating control. A fast floating speed causes excessive overshoot and cycling (provided that a small transfer lag is present) since changes in the flow of control agent are made too rapidly. A slow floating speed allows an excessive deviation of long duration because the control valve does not move at a sufficient rate to check the existing change in controlled variable. The best floating speed is a compromise between excessive cycling and excessive deviation.

FIG. 7-7. Adjustment of Floating Speed for Single-Speed Floating Control with a Load Change.

The greatest advantage of single-speed floating control is that gradual changes in process load can be counteracted by gradual shifting of the valve position. If all lags are small, corrective action can be completed with very little deviation from the neutral. If the load changes suddenly, deviation cannot be avoided. The control valve has only a single speed of motion regardless of the magnitude of deviation, and this speed is generally inadequate for rapidly changing loads.

In addition, single-speed floating control is adaptable to processes having fast reaction rates, where two-position control generally cannot be applied. The greater the process reaction rate, the more successful single-speed floating control can be, especially when transfer lag and dead time are proportionately small.

Summarizing, single-speed floating control is most applicable when:

1. Transfer lag and dead time are small.
2. Process reaction rate is large.
3. Process self-regulation is large.
4. Measuring lag is small.
5. Changes in load are slow.

In many instances where load changes are likely to occur quickly, multispeed floating control can be utilized to limit the deviation. The faster floating speed at the wider limits of deviation produces a faster correction and tends to reduce the amplitude of cycling.

As with two-position control, *immediate reaction of process and controller* is of paramount importance. Under these conditions load changes may not cause excessive deviation.

Single-speed floating control provides the advantage of gradual changes in valve position or flow of control agent. This action is beneficial in the control of furnace temperature where smooth and even firing conditions are sometimes essential.

PROPORTIONAL-SPEED FLOATING CONTROL

Whereas single-speed floating control is generally accomplished electrically, proportional-speed floating control is most often accomplished by pneumatic or hydraulic means. A pneumatic proportional-reset controller can be adjusted to provide essentially the same action as a hydraulic controller by setting the proportional band to a very wide value.
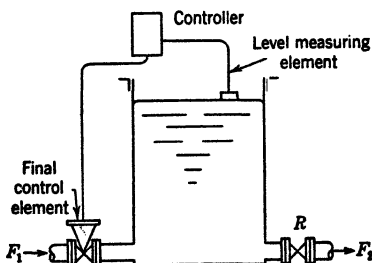
FIG. 7–8. Single-Capacity Process with Automatic Control.

For the *ideal case*, let us suppose that proportional-speed floating control is applied to a single-capacity process, as illustrated by the hydraulic analogy in Fig. 7–8. In this brief analysis it is necessary to assume linear action of the process and valve, and no measuring and controller lags.

The equation for this process is

$$C\frac{d\theta}{dt} = F_1 - F_2 \qquad [7\text{–}1]$$

where C = process capacity.
θ = level in tank.
F_1 = inflow.
F_2 = outflow.
t = time.

With self-regulation in the process, outflow depends upon level, and

$$F_2 = \frac{\theta}{R} \qquad [7\text{–}2]$$

Substituting in equation 7–1,

$$C\frac{d\theta}{dt} + \frac{1}{R}\theta = F_1 \qquad [7\text{–}3]$$

where R is the resistance of the outflow valve.

The relation for proportional-speed floating control is

$$-\frac{dF_1}{dt} = f(\theta - c) \qquad [7\text{-}4]$$

where f is the floating rate of the controller and c is the control point.

Differentiating equation 7–3,

$$C\frac{d^2\theta}{dt^2} + \frac{1}{R}\frac{d\theta}{dt} = \frac{dF_1}{dt} \qquad [7\text{-}5]$$

The controller equation 7–4 can now be substituted in equation 7–5 for the process reaction.

$$(\theta - c)'' + \frac{1}{RC}(\theta - c)' + \frac{f}{C}(\theta - c) = 0 \qquad [7\text{-}6]$$

The primes indicate the appropriate derivatives. The constant c may be arbitrarily added to any derivative, and thus it appears in the derivatives as well. Equation 7–6 represents the action of a single-capacity process with proportional-speed floating control on a change in load (outflow in this example).

We may investigate the nature of this action without directly solving the equation. The magnitude of the coefficients of the equation determines whether the dynamic action is oscillatory or stable. The roots of the equation may be determined from the auxiliary equation, and the imaginary roots may be determined from

$$b^2 - 4ac = 0 \qquad [7\text{-}7]$$

where a, b, and c are the coefficients of each term in equation 7–6.

The stability of the controlled system is excessive when

$$f < \frac{1}{4R^2C} \qquad [7\text{-}8]$$

When the floating rate is slower than the value of the process constants, the action of the system is sluggish, as shown in Fig. 7–9.

The action is critically damped when

$$f = \frac{1}{4R^2C} \qquad [7\text{-}9]$$

The variable then approaches the new balance without overshooting and without excessive stability.

The action is oscillatory but damped when

$$f > \frac{1}{4R^2C} \qquad [7\text{–}10]$$

A slight overshoot is allowed, and the variable oscillates slightly before coming to balance, as shown in Fig. 7–9.
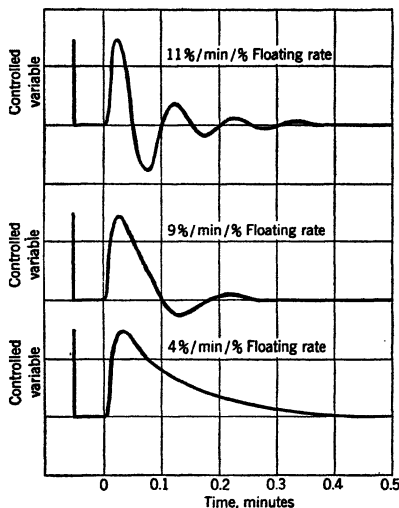
FIG. 7–9. Adjustment of Floating Rate for Proportional-Speed Floating Control with a Load Change.

The control action can be adjusted to be stable by selecting the proper floating rate. A fast floating rate results in excessive cycling before stability is obtained; a slow floating rate produces an overdamped or sluggish control action. Note, however, that proportional-speed floating control does not require a neutral in order to maintain stable action. This arises from the fact that the speed of operation of the valve is dependent upon the deviation of the variable. When the deviation is zero (variable at the control point), the valve is stationary.

With proportional-speed floating control, self-regulation is required in the process in order to achieve stability. If the process contains no self-regulation, equation 7–6 becomes

$$(\theta - c)'' + \frac{f}{C}(\theta - c) = 0 \qquad [7\text{–}11]$$

The outflow is constant, and the second term of equation 7–6 disappears. The solution of equation 7–11 is an equation of simple harmonic motion. The variable simply cycles at constant amplitude. Consequently, self-regulation must be present in order to obtain stable action.

Another characteristic of proportional-speed floating control is that the variable is always returned to the control point for different process loads. In equation 7–6 the rates of change are zero after the variable has become stable and the resultant deviation is zero.

As the process reaction rate becomes larger, proportional-speed

floating control is more effective. This is indicated by equation 7–9, where the floating rate increases as process capacity (C) decreases and as process reaction rate ($1/C$) becomes larger. Figure 7–10 shows the result of using proportional-speed floating control when the process reaction rate is too slow. The period is very much larger. Note also that the floating rate must be decreased when process reaction rate becomes smaller.
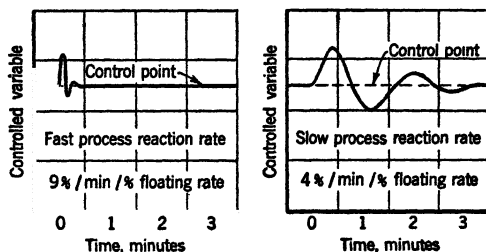
FIG. 7–10. Effect of Process Reaction Rate in Proportional-Speed Floating Control.

In the *practical case*, any appreciable amount of transfer lag or dead time produces an increase in period and a slower floating rate. Consequently, transfer lag and dead time must be small, although a very large or instantaneous process reaction rate will tend to overcome these difficulties.

Proportional-speed floating control does not immediately correct for suddenly changing process loads, although the controlled variable is ultimately returned to the control point. Where appreciable lag is involved, the initial deviation on a sudden load change is more limited by process self-regulation than by control action.

Here, again, a very large process reaction rate may allow a sufficiently fast floating rate so that the controlled variable is quickly returned to the control point.

Therefore, proportional-speed floating control provides the best results when:

1. Transfer lag and dead time are small.
2. Process reaction rate is large.
3. Process has self-regulation.
4. Measuring lag is small.
5. Controller lag is small.

This mode of control is capable of ultimately counteracting load

changes in the process without encountering sustained deviation of the controlled variable. When the deviation is large, a large corrective action is made since the speed of motion of the valve depends upon the magnitude of the deviation. Therefore, proportional-speed floating control maintains a smaller deviation upon load changes than single-speed floating control.

It is important with proportional-speed floating control that *immediate reaction of process and controller* be obtained. Only under these circumstances can the proportional-speed floating mode provide adequate control. Process capacity may then be moderate, and close limits of control may be obtained.

Proportional Control

For investigating an *ideal case*, this mode of control can be applied to the single-capacity process of Fig. 7–8 and the results can be analyzed by the method previously applied.

The controller equation for the proportional mode is

$$-P = \frac{1}{s}(\theta - c) + M \qquad [7\text{–}12]$$

where P = valve position.
s = proportional band.
θ = deviation.
c = control point.
M = manual reset setting.

The process equation is

$$C\frac{d\theta}{dt} + \frac{1}{R}\theta = F_1 \qquad [7\text{–}3]$$

Combining the process and controller equations, when the valve position is synonymous with inflow, and rearranging,

$$(\theta - c)' + \left(\frac{1}{sC} + \frac{1}{RC}\right)(\theta - c) = -\frac{c + RM}{RC} \qquad [7\text{–}13]$$

This equation describes the action of the controlled system, and, without solving it directly, several characteristics of proportional control may be determined.

When the controlled variable ultimately stabilizes, the rate of change is zero, and

$$(\theta - c) \rightarrow -\frac{c + RM}{1 + R/s} \qquad [7\text{–}14]$$

The final steady value depends not entirely upon process load (proportional to R) but principally upon the proportional band setting. Thus, whenever a load change occurs with proportional control, the variable takes a permanent deviation from the control point. This deviation is called offset, and, the wider the proportional band, the greater the offset becomes. By changing the value of M, the manual-reset adjustment may be used to return the variable to the control point, since by equation 7–14, when the deviation $(\theta - c)$ is zero, the manual-reset adjustment (M) must be numerically equal to the ratio c/R.

FIG. 7–11. Adjustment of Proportional Band for Proportional Control with a Load Change.

Offset arises from the nature of the proportional mode of control. In this mode each magnitude of deviation is accompanied by a corresponding position of the valve. When a load change occurs, a greater supply of energy is required by the process. But this additional energy can be supplied only by a greater valve opening, which must be accompanied by further deviation of the variable from the desired point.

The effect of self-regulation, shown in the second term of equation 7–13, is the same as the effect of proportional control. The $1/RC$ term represents the self-regulation; the $1/sC$ term represents the proportional control action.

The effect of process capacity is also shown by equation 7–13. The $1/sC$ term represents the control action. If the process capacity is large, the proportional band must be made smaller if the same absolute value of the sC term is maintained. Smaller proportional bands result in faster corrective action and closer limits of control. Consequently large process capacity is advantageous in proportional control.

In the *practical case,* proportional control may also be applied to a process with more than one capacity where appreciable transfer lag is present. Since most thermal processes have at least two capacities, and the measuring or controller lags further increase that number, proportional control under conditions of transfer lag is a practical problem.

The control action and its stability then depend entirely upon the proportional band setting. Figure 7–11 illustrates the effect of proportional band adjustment upon the recovery of the controlled variable after a change in load in the process. An excessively small proportional band results in excessive cycling before stable action is obtained. It is sometimes possible to decrease the proportional band to a point where the cycling becomes increasingly larger in magnitude and never disappears. A wide proportional band, on the other hand, allows excessive deviation.
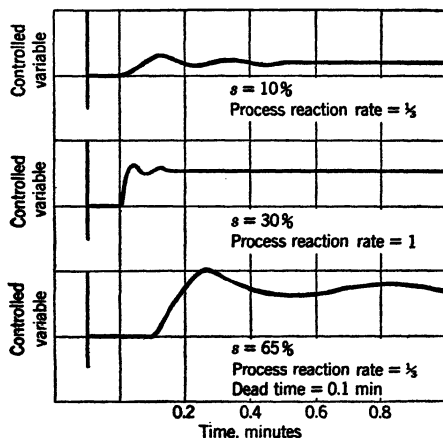
Fig. 7–12. Effect of Process Reaction Rate and Dead Time in Proportional Control.

The variation in offset is also shown in Fig. 7–11. The magnitude of the offset depends upon the proportional-band setting and becomes smaller as the proportional band is made smaller. Since the proportional band is larger for processes having transfer lag, load changes cause proportionately more offset than is desirable.

The process reaction rate in combination with the existing transfer lag or dead time determines, to a great extent, the proportional band. The effect of these factors on the recovery from a load change is shown in Fig. 7–12. The smallest proportional-band setting is brought about by a slow process reaction rate and small transfer lag and dead time. The offset caused by the load change is small.

When the process reaction rate is increased, the proportional band must be widened. The period of cycling, however, is smaller, but the offset is larger. The effect of dead time, as shown in Fig. 7–12, is to

increase greatly the period of cycling and the offset. The proportional band must be greatly increased in order to prevent excessive cycling. It is readily observed that a small dead time causes serious consequences in automatic control.

The characteristics of the proportional mode of control may be summarized as follows:

1. Smaller process reaction rate allows narrow proportional band.
2. Greater transfer lag or dead time requires a wider proportional band.
3. Large magnitude of load change causes permanent deviation of variable.

It should be noted that, if the process has large capacity, small transfer lag, and small dead time, the proportional band which will produce the desirable type of control may be made quite small, sometimes 10 per cent or less. If this is so then the offset accompanying a load change is negligible since the offset cannot generally be greater than a fraction of the proportional band. The magnitude of the load change, then, is less important.

Proportional control, therefore, may be applied to a process when:

1. Large, rapid load changes are not present.
2. The transfer lag and dead time are not too great.
3. The process reaction rate is slow.

With proportional control all three of the above conditions need not be present at the same time since the complete fulfillment of the first or second is generally adequate. For example, if load changes are small or zero, then proportional control with any band setting will result in close control. Or, if transfer lag and dead time are absent, the proportional band will be so small that offset caused by load changes will be negligible.

Use of Reset Response

Two of the modes already described are often combined in order to obtain the advantages of each. They are the proportional mode with its inherent stability, and the proportional-speed floating mode with its stabilization at the control point. The relative amount of each control response is adjustable, and the stability of the controlled variable is dependent upon the adjustment of the controller. The proportional-reset mode of control is the most generally useful of all types of control.

Suppose, for example, that a proportional-reset controller is used
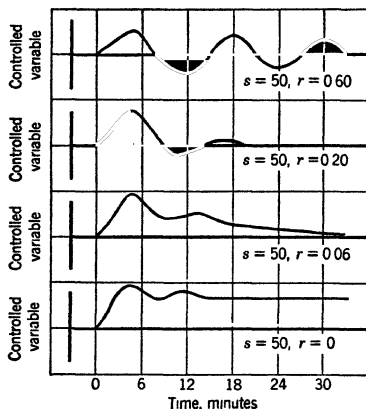
FIG. 7-13. Adjustment of Reset Rate for Proportional-Reset Control with a Load Change.

with a process having moderate reaction rate and transfer lag but little dead time. The control action upon a change in load is shown in Fig. 7-13. Note that the proportional band is set at a constant value for all curves shown. When the reset rate r is zero, the control is accomplished entirely by the proportional mode. The offset of the controlled variable is typical of proportional control.

When the reset rate is set to a small value ($r = 0.06$), the return of the variable to the control point is slow but the offset is ultimately eliminated. This is the purpose of reset response. The recovery curve indicates, however, that the reset rate may be higher.

With the higher reset rate ($r = 0.20$), the return of the variable to the control point is reasonably fast without encountering excessive cycling. This is approximately the optimum setting of reset rate. An even faster reset rate ($r = 0.60$) causes excessive cycling of the variable before a final stable value is reached. Sometimes it is possible to reach a point where the cycling of the variable increases in amplitude and continues indefinitely when the reset rate is made too high. It should be noted that the period of oscillation is comparatively large when it is due to high reset rate.

FIG. 7-14. Adjustment of Proportional Band for Proportional-Reset Control with a Load Change.

The proportional band of a proportional-reset controller may also be adjusted separately. Its effect is illustrated in Fig. 7-14. A change in proportional band also affects the reset response proportionately. For example, if the proportional band is reduced

by one-half, the reset response is increased to twice its former value.

The wide proportional band setting in Fig. 7-14 brings about a slow, gradual return of the variable to the control point. Although the basic reset rate is identical for all curves, the increase in proportional band also decreases the actual reset response until the control action is excessively stable.

Successive narrowing of the proportional band brings about a faster return to the control point and also introduces a cycling action. A proportional band of approximately 50 per cent is optimum in this particular example. A narrower proportional band produces excessive cycling so that a long time elapses before the controlled variable is stabilized at the control point. The period of cycling is much smaller in comparison to the cycling caused entirely by reset response shown in Fig. 7-13. This is one method by which the cause of cycling can be identified.

Smaller process capacity permits a higher reset rate when there is little transfer lag or dead time. This corresponds to a higher floating rate with a proportional-speed floating controller. Transfer lag or dead time in the controlled system, however, requires that the reset rate be considerably decreased in order to avoid excessive cycling of the controlled variable.

The influence of the characteristics of the controlled system on the relative adjustment of each response in a proportional-reset controller is summarized in the table below.

Process Characteristic	s, Proportional Band	r, Reset Rate
Small N, large D	Moderate	Slow
Small N, small D	Narrow	Fast
Large N, large D	Wide	Slow
Large N, small D	Moderate	Fast

N is process reaction rate; D is transfer lag or dead time.

The proportional-reset mode may usually be applied to any process having characteristics suitable for the application of either mode separately. In addition, there is no requirement for self-regulation in the process as with proportional-speed floating control. The proportional response is sufficient to provide the necessary stability.

The limitation of applying proportional-reset control lies only in the

large period of the controlled system and the attendant slow response when dead time is present. Transfer lag is not a limitation as long as the measuring means is sensitive to every small change. Otherwise, the transfer lag is converted to a dead time, the proportional band is made higher, and the reset rate is made slower.

Consequently we find that the major controlling effort can be made by either the proportional or the reset control responses. For processes with small lags of any kind the control is accomplished mainly by the action of reset response. For processes with large capacity and small or moderate transfer lag and dead time the main control effort is provided by the proportional mode.

If the proportional mode provides the major part of the controller action, then the purpose of the reset response is to accomplish a slow movement of the valve to eliminate offset of the controlled variable. The effect of the proportional response is generally more stabilizing than the effect of the reset response.

In the control of processes with little capacity but no transfer lag or dead time the principal part of the controller action is usually provided by the reset response. The combination of the proportional mode with the proportional-speed floating mode is beneficial in spite of the fact that only a wide proportional band can be used. The improvement in control is due to the stabilizing effect of the proportional response.

Use of Rate Response

Processes having large dead time, or large transfer lag, may sometimes prove difficult to control. Large lags of these kinds practically preclude the choice of any mode of control except proportional-reset. The proportional band must be exceptionally wide and the reset rate very slow in order to avoid continuous cycling. When load changes occur, excessive deviation is obtained and a longer time is required to recover from the change.

The application of rate response for counteracting the effect of dead time is shown in Fig. 7–15. The addition of rate response to the proportional-reset controller substantially reduces the maximum deviation. The most notable improvement is produced in the period of cycling, which is reduced by about one-half. Thus the controlled variable is stabilized at the control point in about one-quarter of the time required without rate response.

The adjustments of the controller generally require substantial change of settings when rate response is used. In Fig. 7–15, the proportional band (s) was reduced to about one-half and the reset rate (r)

was considerably increased. The resultant effect is to produce much faster corrective action, as evidenced by the recovery curves.

The most serious aspect of transfer lag is that the first indications of changes in process balance are small in magnitude and occur at a slow
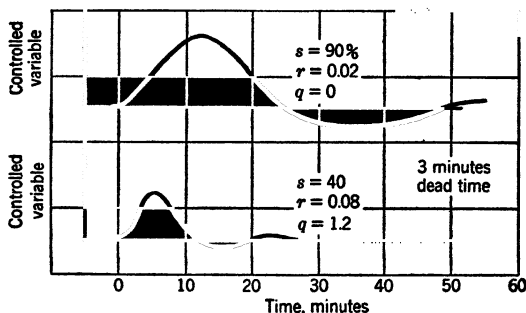
Fig. 7-15. Proportional-Reset versus Proportional-Reset-Rate Control Where Transfer Lag and Dead Time Exist.

rate. If the measuring means possesses a dead zone, a dead time results. The use of rate response may allow satisfactory control in spite of any dead zone in measurement. The presence of large measuring or controller lags produces the effect of transfer lag because of
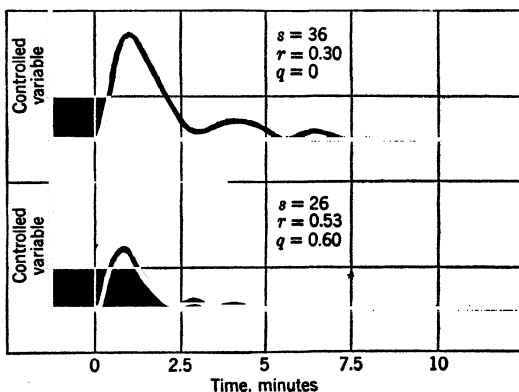
Fig. 7-16. Proportional-Reset versus Proportional-Reset-Rate Control Where Transfer Lag but No Dead Time Exists.

the increased number of capacities in the controlled system. Under such conditions, rate response may be employed to counteract the effect of the lags.

The effect of rate response in counteracting transfer lag is shown in Fig. 7-16. No dead time is present in this example. Although con-

siderable improvement in the recovery curve is indicated, the results might possibly have been further improved by a readjustment of the controller. The period of cycling is reduced by about one-quarter and the controlled variable is stabilized more quickly. Rate response is effective when transfer lag exists but is more generally applicable for combating dead time since it is the only means for doing so.
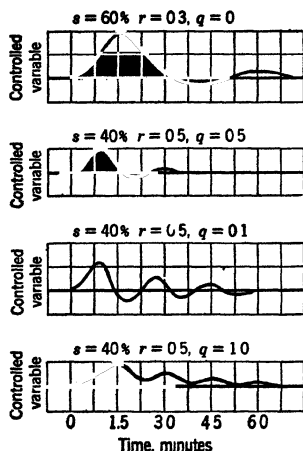
FIG. 7-17. Adjustment of Rate Time for Proportional-Reset-Rate Control with a Load Change.

The adjustment of rate time for rate response must be made carefully since a large rate time causes excessive oscillation much the same as a small proportional band or large reset rate. Figure 7-17 illustrates this action. When the rate time (q) is increased beyond the proper amount, cycling of the controlled variable is caused. Notice that the period of cycling is comparatively short. A large setting of rate time opposes any fast change in the controlled variable either away from or toward the control point.

When the rate time (q) is readjusted to a low value, cycling is also caused, but it is due to the small proportional band and larger reset rate which had previously been satisfactory with rate response. The period of cycling is noticeably larger than the period caused by too large rate time. This is a means of recognizing the cause of excessive cycling. A cycle caused by reset response is long; by proportional response, short; and by rate response, even shorter.

The curves of Fig. 7-17 point to the stabilizing influence of rate response. The smaller proportional bands that are possible with rate response are directly due to this stabilizing effect. After the proportional band has been reduced, an appreciable decrease in rate time may produce cycling because of the narrow proportional band and the absence of the stabilizing influence of rate response.

Because of the stabilizing influence of rate response, sudden load changes can be more quickly counteracted. With rapid load changes it is possible to adjust the rate time to a maximum so that corrective action starts before the deviation has become excessive. When dead time is large, however, rate response is less effective in limiting maxi-

mum deviation than it is in providing a rapid return to the control point.

Summary of Modes of Control

Generally each of the modes of control is applicable to processes having certain characteristics; the importance of fitting the mode of control to the process must not be overlooked.

In automatic control, the absence of measuring lag, controller lag, transfer lag, and dead time permits the choice of any mode of control. Deviation of only a few tenths of 1 per cent of scale is possible under these ideal circumstances. The difficulty lies in finding a controlled system without any of the above-mentioned four lags.

The complete absence of transfer lag and dead time permits the choice of either two-position, single-speed floating, or proportional-speed floating mode of control. In addition, two-position control operates best with slow process reaction rate. Floating control requires self-regulation in the process and a fast process reaction rate.

Proportional control requires moderate process reaction rate and small load changes in order to minimize offset. Proportional-reset control may be suitable under any conditions of process reaction rate since one of the responses applies to either small or large process reaction rate. The addition of rate response will permit closer control under conditions of dead time or sudden load changes.

The table below summarizes the principal characteristics of the controlled system for each mode of control. Naturally, the table is useful only for broad classification of control systems.

Mode of Control	Process Reaction Rate	Transfer Lag or Dead Time	Process Load Changes
Two-position	Slow	Slight	Small and slow
Single-speed floating*	Fast	Slight	Slow
Proportional-speed floating*	Fast	Slight	Slow or moderate
Proportional	Slow or moderate	Small or moderate	Small
Proportional-rate	Slow or moderate	†	Small
Proportional-reset	†	Small or moderate	Slow
Proportional-reset-rate	†	†	†

* The process must have self-regulation for floating control.

† In general, any amount

It is very often possible to exceed the limits shown in the table above. For example, if transfer lag and dead time are zero, almost any mode of control may be successful. Many laboratory processes, such as jacket temperature control, can be designed so that the total variation in temperature is only a few hundredths of a degree, even under severe load-change conditions. The average industrial control application, and many domestic systems, will, however, fit the table.

Theories for Quantitative Analysis

The study of automatic control by pure mathematics alone, without coordinating tests, is generally complicated even when assumptions are made in order to simplify the problem. The usual method is to convert the process under control to some simple analog, either electrical, hydraulic, thermal, or mechanical, and determine the reaction of the controlled system from a study of the analog. The results can then be transferred into terms of the actual process.

The number of capacities of an industrial process determines the order of the differential equation describing it. If the process has no capacity, the minimum number of capacities is two, one each for the controller and measuring lags, and a second-order differential equation is indicated. If the process has two capacities, the measuring lag two, and the controller lag one, the differential equation is of the fifth order. This condition is the more usual.

It is also necessary to assume that the differential equations have linear coefficients; otherwise solution must be made by approximation methods. Non-linearities are introduced by changing resistances and capacities in the process, by radiation and flow which are power relationships, and by valve characteristics. Non-linearity may not handicap the solution if the operating characteristics at one point are known and if the solution is restricted to that point.

Several methods of mathematical analysis which have been employed will be presented briefly.

Ivanoff[9] followed a method in which it is assumed that the measured variable without control will change its magnitude in a definite manner, indicated by what we have here called the reaction curve. If the variable is temperature the uncontrolled value is termed the potential temperature. When automatic control is applied mathematically the potential magnitude of the controlled variable is governed so that the actual magnitude of the controlled variable becomes stable.

Callendar, Hartree, Porter, and Stevenson[2, 7] described a method of mathematical analysis in which the differential equation for the controller is modified by the inclusion of the process characteristics of

reaction rate and dead time. Solutions are obtained, and a graphical relation is given between the various characteristics of the controlled system. An analysis of rate response and reset response, and the reaction of a controlled system to various types of load changes, are included in the articles referred to.

Mason and Philbrick[11] and Spitzglass[17] employ direct mathematical means for setting up the differential equations for the process. The controller is then applied to the process, and the differential equation for the controlled system is solved for various stability conditions. This general method in its simplest form has been followed in the present and the preceding chapters for analyzing characteristics of processes and automatic control systems.
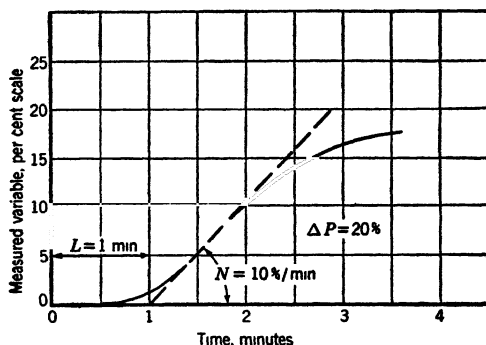
FIG. 7-18. Approximation of Process Reaction Curve. (From Ziegler and Nichols, *ASME Trans.*, reference 19.)

Liu[10] developed a means of analyzing stability and transient effects in controlled systems. It is first necessary to determine the type of differential equation describing the controlled system. Without solving the equation, the stability of the controlled system may be determined directly from a set of stability graphs which apply to a particular degree of system.

Hall[6] has greatly extended the transfer loci method which was described by Nyquist.[13] Although this method is developed for application to linear servomechanisms, it may also be applied to automatic control. The method consists of determining the constants for the particular differential equation describing the controlled system, and determining the stability of the system by means of a graph which can be quickly plotted for each system.

The Ziegler-Nichols method,[19, 20] which will be described in more

detail, consists of solving directly for controller requirements from empirical equations. Quantitative analysis of the controlled system by these means makes possible the selection of the mode of control.

The reaction curve for the controlled system, as shown in Fig. 7–18, is derived from the characteristics of the process, the measuring means, and the controlling means. Usually enough accuracy can be obtained with the design data for the process. Thus the reaction curve can be derived without having the actual process at hand. The reaction curve can, of course, be obtained by test of the actual controlled system.

The reaction curve can be approximated by a *reaction rate* N and a *lag* L. A line drawn tangent to the point of inflection of the curve intersects the time axis at a distance from zero. The time shown on the curve is called the lag L of the controlled system. The slope of the line is the maximum rate of change of the measured variable and is called the reaction rate N of the controlled system. The per cent change in valve or other final element position to produce the reaction curve is called ΔP.

Several values of these factors have been determined for various applications and are listed in the table below.

Process	Controlled Variable	Change ΔP	Reaction Rate N	Lag L
Ammonia absorber	Temperature	83	16	8.7
Fractionating column	Temperature	42	6.3	7.5
Wet bulb	Temperature	50	2.5	4.5
Oil-tube still	Temperature	25	8.2	3.1
Superheater	Temperature	25	24	2.1
Dry bulb	Temperature	16	9.3	0.77
Milk heater	Temperature	4	25	0.67
Water flow	Flow	33	47	0.12
Column vent	Pressure	4	2.1	0.08
Canning retort	Temperature	17	7.2	0.03

The lag factors in the table give a particularly good picture of the wide variations encountered in controlled systems. It must be noted that the data are representative of one specific application and should not be assumed correct for other applications of similar nature.

The adjustments for the proportional mode can be estimated by solving a set of empirical equations. These relations are derived from a study of the stability of controlled systems.

For the proportional mode of control the proportional band can be determined from

$$s = \frac{100NL}{\Delta P} \qquad [7\text{–}15]$$

where s = proportional band in per cent.
N = reaction rate in per cent of scale per minute.
L = lag (from reaction curve) in minutes.
ΔP = per cent of change in the valve position required to produce the reaction curve.

For the proportional-reset mode of control the adjustments can be determined from

$$s = \frac{110NL}{\Delta P} \qquad [7\text{–}16]$$

$$r = \frac{0.3}{L} \qquad [7\text{–}17]$$

where r is the reset rate. For the reaction curve of Fig. 7–18 the proportional band should be 55 per cent and the reset rate 0.3 per minute.

For the proportional-reset-rate mode of control the adjustments can be determined from

$$s = \frac{83NL}{\Delta P} \qquad [7\text{–}18]$$

$$r = \frac{0.5}{L} \qquad [7\text{–}19]$$

$$q = 0.5L \qquad [7\text{–}20]$$

where q is the rate time. For the reaction curve of Fig. 7–18, the proportional band would be 41 per cent, the reset rate 0.5 per minute, and the rate time 0.5 minute.

Obviously, if after solving the above equation the proportional band is found to be less than 3 or 4 per cent, two-position control may be possible. It is an interesting problem to solve the equations for the processes listed in the table above.

The Ziegler-Nichols method has limitations common to most empirical methods. The approximation of a reaction curve by a time lag and a straight line is not invariably valid. For example, if the process reaction curve indicates large transfer lag the difference between the settings required by a relatively sensitive and an insensitive measuring means may be quite significant.

This method does not provide the optimum settings for all types of load changes but for only the one particular type used in determining the reaction curve. This is hardly serious, since the only requirement is that the controller be adjusted for the worst conditions. This

method, however, is one of the simplest means of analyzing processes with dead time.

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Chapter 8

QUALITY OF AUTOMATIC CONTROL

The results of automatic control must always be evaluated in terms of the quality of the finished product rather than in terms of accuracy or deviation of the controlled variable. The general purpose of automatic control is to obtain maximum efficiency of process operation. Efficiency of operation, however, may be obtained in an entirely different manner in different processes.

A number of factors affect the quality of control, the most important of which is the behavior of the process load. The process load may be defined as the sum, taken at any one time, of the requirements of the process for flow of control agent. It is easily expressed in terms of the setting of the final control element to maintain the controlled variable at a given point.

Changes in process load cause deviation of the controlled variable. The rate at which the process load changes governs the magnitude of deviation to a great extent. The magnitude of the process load is a large factor in determining the process reaction. The action of the controlled variable under dynamic conditions of load change is called the recovery. Controller adjustments are selected to produce the desired stability of control under the existing magnitude of load changes and rate of load changes in the process.

Another factor affecting the quality of control is the systematic response of both the process and the control system. The action of a controlled system depends upon the precise and consistent response of the controller to changes in the controlled variable, and upon the precise and consistent response of the controlled variable to changes in the control agent. If these responses change with time, the controller is greatly handicapped in performing its function properly.

Origin of Load Changes

Since the process load is defined in terms of the energy requirement of the process, a change in magnitude of any of the variables associated with the balance of the process constitutes a load change. The controller must then readjust the energy supply to the energy loss in order to maintain the desired magnitude of the controlled variable.

To illustrate, suppose that the temperature of the gas-fired continuous heating furnace of Fig. 8–1 is to be controlled to a constant value. The heat losses are made up by stack, radiation, and work. The heat supply is made up by work and fuel. If the sum of the heat supply is equal to the total heat loss the temperature must be constant and unchanging.
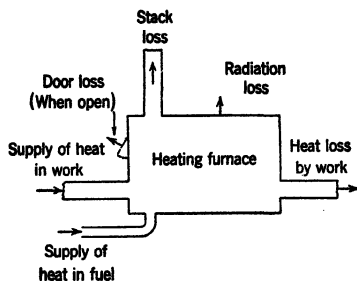
FIG. 8–1. Location of Heat Supply and Loss in a Thermal Process.

When the door is opened an additional loss is created and the temperature drops. The extra loss must be made up by a greater supply of heat. The controller readjusts the valve setting to a greater opening to return the temperature to the control point. In this example the load has changed and the controller is relied upon to maintain the controlled variable at the desired magnitude.

A load change is the result of a number of circumstances. The first of these has already been illustrated in the form of a change in an uncontrolled condition, the opening of the furnace door. These changes are found in the uncontrolled variables associated with the process. In the example of Fig. 8–1, load changes may be caused by a change in stack loss due to changing draft or by radiation loss due to varying ambient conditions at the furnace.

A second type of load change, caused by variations in demand of energy by the material being processed, is called a demand change and is illustrated in Fig. 8–1 by a change in the supply of heat by the work. The heat supply in the work may be changed by varying the temperature of the work as it enters the furnace. The most common demand change, however, is caused by variations in the rate at which the work flows through the process. A greater amount of work in the furnace at any one time or during any period requires a greater firing rate to attain the desired temperature of the metal.

A third type of load change is caused by variations in the rate of energy supply due to causes other than controller action. For example, if the gas pressure changes in Fig. 8–1, the flow of fuel will be altered. Another example is the clogging of the burners which decreases the flow of fuel for any one valve setting. Correction for these changes in supply must be made by the controller.

A fourth type of load change is caused by changing the control point of the controller. In the example given, the control point may be raised to a higher temperature. The energy requirements become

greater and the supply of heat must accordingly be made greater in order to balance the process at the new set of conditions.

The types of load changes are, therefore:

Changes in associated variables.
Changes in demand.
Changes in supply.
Changes in control point.

Supply changes cause a different reaction from those caused by changes in demand. The process reaction curves of Fig. 8-2 illustrate this point. If the measuring lag is small, a demand change always causes this typical single-capacity reaction. Note that a demand change is reflected in the controlled variable quickly and with relatively large initial magnitude. A supply change, however, involves a delayed reaction in the measured variable, and the initial magnitude is much smaller in comparison with the demand change. If the process transfer lag is large, the effect of a supply change is even more greatly delayed.
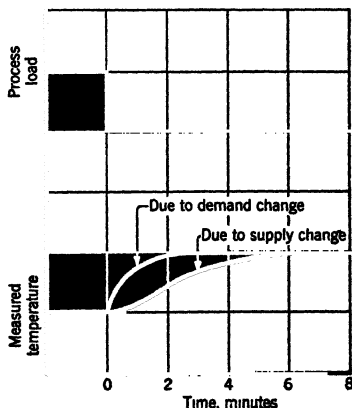
FIG. 8-2. Effect of Uncontrolled Process Load Changes.

Suppose that a proportional-reset controller is applied to a process in which load changes of the types listed above can be made. The actual adjustment of the controller is not important for this discussion. A change in supply causes a deviation and recovery of the controlled variable as shown in Fig. 8-3. The change in controlled variable is not detected by the controller until it has passed completely through the process. Corrective action does not begin immediately, owing to the transfer lag and dead time in the controlled system.

A load change caused at some intermediate point in the process, but neither a supply nor demand change, affects the controlled variable more rapidly since it is delayed by only the part of the process between the point of the change and the primary measuring element.

A demand change results in almost immediate reaction of the controlled variable and is delayed only by the extent of the measuring lag. The recovery curves indicate that a demand change can be stabilized more quickly than a supply change. The total disturbance time for

each type of load change depends to a great extent on the arrangement and number of capacities in the process.

If the control point of the controller is suddenly shifted to a new value, the controlled variable must stabilize at a new point as shown in Fig. 8–3. Under these conditions the position of the final control element is changed at the same time that the control point is shifted. The resulting recovery curve is very similar to the recovery curve for a supply change in that the reaction must pass completely through the process.
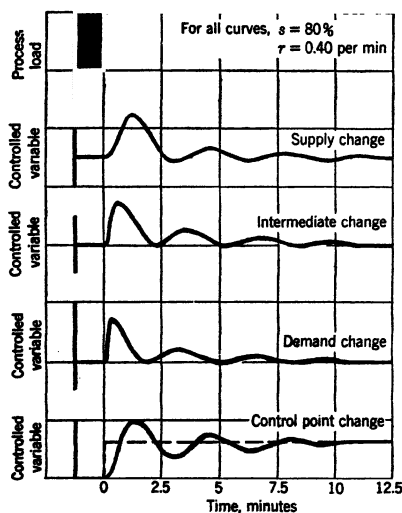
FIG. 8–3. Recovery Curves for Various Process Load Changes.
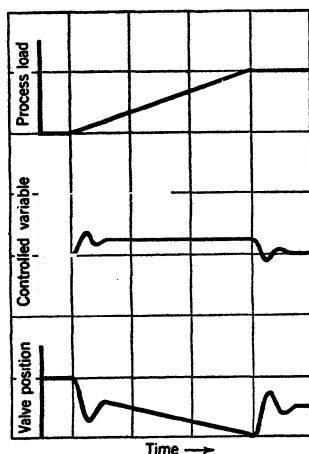
FIG. 8–4. Effect of Constantly Changing Process Load with Proportional-Reset Control.

RATE OF LOAD CHANGE

A characteristic of greatest importance to quality of control is the rate at which process load changes take place. Load changes do not usually occur instantaneously or with such suddenness as assumed in Fig. 8–3. Ordinarily process load changes are gradual and have a finite rate of change.

The effect of a slow instead of an instantaneous load change is shown in Fig. 8–4. Neglecting transient effects, it is necessary for a constant deviation to take place in order that the control valve be given a gradual change in position. This action occurs automatically with a proportional-reset controller. The deviation of the controlled variable balances

at a magnitude sufficient to enable reset response to shift the valve at a rate commensurate with the rate of load change.

If the rate of valve motion required to correct for the gradually changing load is known it is possible to calculate the deviation which would result. Since reset response provides the corrective action,

$$\frac{dP}{dt} = \frac{r}{s}(\theta - c) \qquad [8\text{-}1]$$

where P = valve position,
t = time,
r = reset rate,
s = proportional band,
$(\theta - c)$ = deviation of controlled variable,

and equation 8-1 is one term of the equation for the proportional-reset controller.

Solving for deviation,

$$(\theta - c) = \frac{s}{r}\frac{dP}{dt} \qquad [8\text{-}2]$$

Fig. 8-5. Effect of Rate of Load Change with Proportional-Reset Control.

Thus, as long as the reset rate can be made large and the proportional band small, the deviation caused by a constantly changing load may be made quite small.

The effect of different rate of load changes is illustrated in Fig. 8-5. A slow rate of load change results in a small deviation, since only a slow motion of the control valve is required to supply the changing demand. A fast rate of load change, neglecting transient effects, causes a larger deviation if the controller adjustments are the same. As the rate of load change becomes even faster, the transient conditions (sudden surges in the controlled variable) become more important since the initial deviation is generally increased. A sudden or instantaneous load change produces a transient recovery with no persistent deviation if reset rate is properly set.

In considering the effect of rate of load change, a proportional-reset controller has been assumed since reset response is nearly always required under conditions of changing process load. The proportional-speed floating controller reacts in exactly the same manner. The proportional controller can provide the gradual change in valve position only by an increasingly greater deviation until the full load change and full offset are obtained.

A large and sudden load change is an unusual but particularly difficult characteristic in automatic control; process self-regulation may not be sufficient to limit the deviation to a reasonable value, and controller action must be relied upon to produce effective corrective action.

A small proportional band is effective in correcting for large sudden deviations. However, if the transfer lag or dead time is sufficient, it may not be possible to reduce the proportional band to a point where proportional action is effective. Rate response may sometimes be applied so that faster corrective action may be instituted when sudden deviation begins. Response to rate of change of the controlled variable is applicable when a sudden load change results in a sudden change in the controlled variable. Otherwise, changes which occur during a dead-time period are inevitable, and it is generally better to correct the load change at its source whenever possible.

Magnitude of Process Load

Often a controller is handicapped by being called upon to operate under conditions of large variations in process load. The resultant effect upon the quality of control may be such that a controller satisfactorily operating with high process load may produce entirely undesirable results at low process load. In a majority of applications, the process load does not change over such a wide range, and small differences in process reaction do not affect the control results.
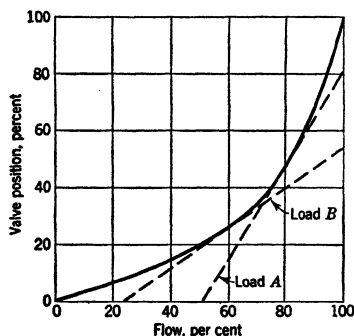
Fig. 8–6. Relation of Valve Characteristic to Process Load.

Three major characteristics of the controlled system are related to process load: the characteristic law, usually a lift-flow relationship, of the final control element; the process load-reaction rate relationship; and the controller scale characteristic. It will not be possible to describe all the combinations of these factors; the effect of each will be only generally indicated.

In analyzing the effect of the final element characteristics it is necessary to assume a linear controller scale and a constant process reaction rate. Suppose, for example, that the flow characteristic of the final element or control valve in a controlled system is similar to Fig. 8–6. If the process load is of such a magnitude that an 80 per cent flow of

control agent maintains the variable at the control point, then the controller may be considered to operate about the dotted line designated load A. If the proportional band of the controller is set to 40 per cent, a deviation of the controlled variable of 1 per cent results in a 1.5 per cent change in flow.

A decrease in process load of any appreciable magnitude, however, shifts the operating point of the final element and causes a change in the characteristics of the system. If the new load is at load B in Fig. 8-6, a change in the resultant effect of the proportional band is produced. A deviation of the controlled variable of 1 per cent now results in a 3.5 per cent change in flow. The controlled system will now react as though the proportional band had been reduced to 17 per cent when it has not actually been changed.

If the controller has been set to produce a desired recovery curve at load A, it will undoubtedly cycle excessively at load B. If the cycle is to be eliminated, the proportional band must be increased to 90 per cent, in this example, in order to return to the type of recovery originally selected.

Although this example is exaggerated in order to illustrate the effect of the magnitude of the process load, the control system may sometimes need special consideration from this standpoint. The actual characteristic of the final control element, *from consideration of controller and final element only*, should be linear; that is, it should provide equal increments of flow for equal increments of change in position. A linear characteristic always results in a constant overall proportional band at all average positions of the final element.

If the controller can be set to a small proportional band by virtue of small transfer lag, small dead time, and small process reaction rate, the effect of changing process loads may be averaged by properly selecting the adjustment of the controller. A slightly too wide proportional band at high load may only result in a slightly too narrow band at low load.

The magnitude of the process load also influences the process reaction in many applications. We may investigate this part of the problem by simple calculation of a process analog similar to that shown in Fig. 8-7. The constants for this process may be made the same as those for the hydraulic analogy which was analyzed in a preceding chapter. Translated into thermal units,

$$\text{Inflow } F_0 = 6.5 \text{ lb water/min}$$
$$\text{Temperature at } F_0 = 78° \text{ F}$$
$$\text{Outflow } F_2 = 6.5 \text{ lb water/min}$$
$$\text{Temperature at } F_2 = 140° \text{ F}$$
$$\text{Flow } F_1 = 400 \text{ Btu/min}$$

The reaction curve for the process *at high load* may be determined by reducing flow F_1 to 270 Btu per minute. The reaction curve for the process is shown in Fig. 8–7.

Variation in the process load produces a decreased requirement for the heating medium at F_1. Suppose, for example, that the flow of heated water at F_0 is reduced. The new conditions may be represented by

$$\begin{aligned} \text{Inflow } F_0 &= 3.7 \text{ lb water/min} \\ \text{Temperature at } F_0 &= 78^\circ \text{ F} \\ \text{Outflow } F_2 &= 3.7 \text{ lb water/min} \\ \text{Temperature at } F_2 &= 140^\circ \text{ F} \\ \text{Flow } F_1 &= 230 \text{ Btu/min} \end{aligned}$$

The reaction curve for the same process *at low load* is determined by reducing flow F_1 to 155 Btu per minute. This reaction curve is also shown in Fig. 8–7. It is immediately obvious that the process has a substantially different reaction at different process loads.
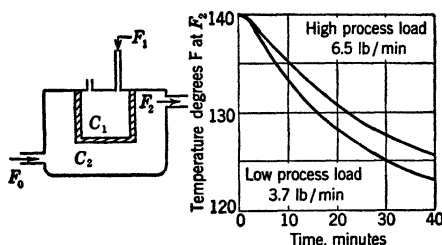
Fig. 8–7. Effect of Process Load on Process Reaction Curve.

In order to investigate the effect upon the quality of control, we may apply the Ziegler-Nichols method for determining the proportional band required for a proportional controller. Regardless of whether this method provides theoretically absolutely correct results, it will permit an easy solution of this problem.

By calculating the process reaction at both high and low loads, assuming a control point of 130° F, a valve size for use at both high and low loads, and a controller scale calibration, it is possible to determine the process reaction rate and the process lag. These results are as follows:

Process Load, lb/min	Process Reaction Rate, %/min	Process Lag, min	Inflow Change, %
6.5	3.42	1.09	26.2
3.7	2.26	1.34	15

By employing the relation between lag, reaction rate, and controller proportional band, the controller should be set as shown below. An additional lag of 0.5 minute is assumed to be provided by the controller and measuring lags.

High process load (6.5)	Band = 20
Low process load (3.7)	Band = 28

The result of the change in process load in this example is to require an appreciably larger proportional band at low load than is required at high load. In some applications where process load changes are large, it may be necessary to widen the proportional band at low process loads in order to avoid excessive cycling.

The effect of the characteristic of the final element on the actual proportional band at different process loads has been previously investigated. It is generally possible to combine these characteristics so that they will at least partially compensate each other.

One type of flow characteristic of final elements, particularly control valves, available for this purpose has a semi-logarithmic flow-lift relationship such as shown in Fig. 8–8. Suppose that this type of control valve is selected for the process of Fig. 8–7. The average slope at the two operating loads is indicated by the tangents to the curve of Fig. 8–8.
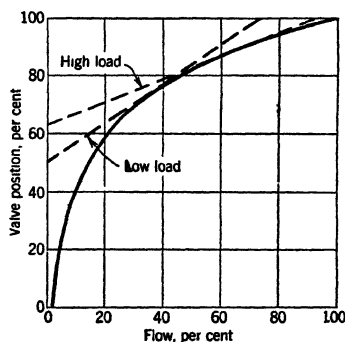
Fig. 8–8. Use of Valve Characteristic for Compensating a Changing Process Reaction Rate.

By using the change in slope of the characteristic at different valve positions, the proportional band at the controller may remain fixed while the actual resultant proportional band varies with process load. This is indicated below in the table.

Process Load, lb/min	Controller Band Setting, %	Actual Band Desired, %	Actual Band with Semi-Log Valve, %
6.5	46	20	18
3.7	46	28	30

This method of compensating for changing process characteristics at various loads results, in this example, in a fixed magnitude of controller band setting while the actual band varies with process load.

The difficulty in applying this method for every control application is

caused by two important factors: first, changes in process reaction with load are different for every process; and, second, as previously emphasized, valve flow and area characteristics are not always identical. Consequently, from a theoretical standpoint, different valve characteristics are required for different process arrangements. Valves with adjustable characteristics are not readily obtainable, and the semilogarithmic and parabolic valve characteristics are in general use for this purpose. It must be emphasized, however, that process loads do not often vary to any great extent so that effects of process reaction and load are minimized.

Controller scale characteristic may be another variation requiring consideration when controller scale calibration is non-linear, such as with some flow controllers or vapor-actuated thermometer controllers, and when the control point is shifted to different positions on the scale. The effect of this characteristic may be calculated like that of varying process reactions, by determining the various valve positions required to maintain the controlled variable at the control point and proceeding as before.

Stability of Control

The adjustments provided on proportional-speed floating, proportional, proportional-reset, and proportional-reset-rate controllers permit a selection of the action of the controlled variable in recovering from a
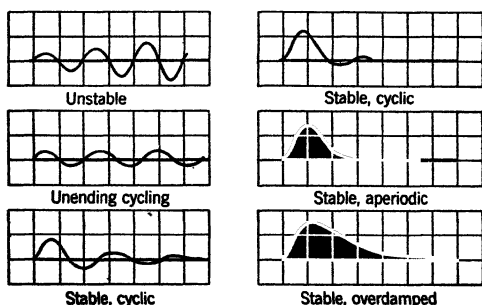
Fig. 8-9. Stability of Control.

load change. The stability of control as illustrated by the recovery is defined as the property of the combined effects of control system and process upon one another, whereby the controlled variable is maintained within limits without sustained cycling.

The characteristic action of the controller in performing its service of

automatic control is important simply because use may be made of each type of stability. Two-position control, being different from any other mode of control, results in only a single kind of stability — unending cycling. Most other types of control may produce other kinds of stability under widely varying conditions of process lag and controller application.

The five types of stability are shown in Fig. 8–9. Unstable action results from what is commonly termed "too strong" control action — where proportional band is too small, reset rate is too large, or, sometimes, rate time is too large. Overdamped action results from "too weak" control action — wide proportional band, slow reset rate, or, sometimes, large rate time.

The criteria by which the stability of control is usually judged are minimum area, minimum deviation, and minimum cycling. It is generally possible to adjust the controller to obtain the average stability of control nearest to one of these three requirements.

Fig. 8–10. Criteria for Stability of Control.

Minimum area under the deviation curve is desirable for most average conditions where no special considerations are involved. Figure 8–10 illustrates the recovery curve of the controlled variable upon a load change. It must be noted that, in proportional control, offset is obtained and the controlled variable lines out away from the control point. When the shaded area between the control point and the deviation curve is a minimum, the deviation averages the smallest amount for the shortest length of time. At this stability the amplitude ratio (amplitude of each cycle divided by amplitude of the preceding cycle) should be about 0.25.

Recovery with minimum area applies to the control of processes in which time of departure from the control point is of as great importance as magnitude of deviation. In order to stabilize the controlled variable in the shortest time, optimum stability must be obtained. This is practically synonymous with smallest area of deviation. In this way the smallest quantity of product is the least improperly processed. For

example, in continuous food processing, any deviation of temperature below the control point may result in undercooking. A deviation above the control point may result in overcooking. As long as the deviation remains within reasonable limits, minimum area of recovery results in the highest quality of product.

Minimum deviation is a requirement for recovery of many controlled systems where ill effects are caused in the process equipment or in the product by momentarily excessive deviation. Here the magnitude of deviation is of greater relative importance than time of deviation. The limits of control are therefore generally narrow, and it is desired to remain as close as possible to the control point.

A recovery with minimum deviation as shown in Fig. 8–10 might be required in a thermal oil-cracking furnace, for example. In this application, even very small deviations will cause large changes in the rate of cracking. Furthermore, higher temperatures result in carbon deposits inside the heat exchanger tubes and production is sometimes materially decreased. Minimum deviation is gained at a sacrifice of smooth control and greater area between the control point and deviation curve is obtained.

It should be noted that while all the curves of Fig. 8–10 are drawn from tests using proportional-reset mode of control for each example, process self-regulation is an important factor in reducing the maximum deviation obtained upon a change in load. Obviously if the process has comparatively large self-regulation a change in load causes less deviation of the controlled variable. The large increase in energy transfer resulting from self-regulation may be sufficient to limit the deviation until corrective action by the controller can be made fully effective. This is especially true when dead time is present, since corrective action cannot be effective until the dead-time period has passed.

Minimum cycling is sometimes required where periodic disturbances in the processing unit and plant must be avoided. If the output of one process is carried to a second process, as often occurs in continuous chemical processing, any cycling of the controlled variable becomes a cycling load change to the second process. Greater stability of control can be obtained and cycling kept at a minimum with a recovery as shown in Fig. 8–10.

Greater stability is gained by sacrificing both minimum deviation and minimum area. The recovery can be critically damped without cycling or overshooting, and the return of the controlled variable to the control point is essentially asymptotic.

All the criteria for judging stability of control are derived from the purpose and requirement for which automatic control is used. *The*

quality of control is, therefore, always relative in the sense that it must necessarily depend upon factors of application. A satisfactory recovery and stability in one controlled system may be entirely unsatisfactory in another.

Adjusting Controller Response

Adjustment of the controller, whether it be two-position, floating, or proportional in action, is necessary in order to obtain most efficient operation of the process. It is a point too likely to be overlooked when partially satisfactory results are obtained from haphazard selection of adjustments.[1] The quality of control may generally be improved upon by using a logical method for selecting controller adjustments.

The stability of control and recovery of the controlled system are, in a large measure, dependent upon the adjustment of the controller. Usually the characteristics of the process and complete controlled system can be studied more readily from observation of the settings of a proportional-reset controller than by any other method since controller settings for a properly adjusted controller may usually be interpreted in terms of process lags.

FIG. 8-11. Effect of Size of Final Element in Two-Position Control.

During the adjustment of controllers it is important to allow sufficient time for the effect of each adjustment to be observed. Very often an unsatisfactory quality of control results from adjustments selected on the basis of incomplete data. On processes having slow reaction rates or very long periods it may be necessary to wait for hours before a complete recovery curve can be obtained. There is no substitute for time spent in properly adjusting a controller, and returns in the form of efficient process operation are usually high.

Two-position controllers are generally not adjustable to process characteristics, but they thereby gain the advantage of simplicity. One of the important factors in the operation of a two-position controller is the size of the valve or other final control element. An oversize element may result in greater amplitude of cycle of the controlled variable than is necessary. The excessive rate of change of the controlled variable causes the variable to overshoot the control point as shown in Fig. 8-11.

The cycle then becomes unsymmetrical about the control point, and the period is lengthened.

If the final control element has a range adjustment, or minimum and maximum flow adjustments, the range of operation may be limited and the cycle of the controlled variable made more even. Not only is the quality of control improved but also less fluctuation of the supply prolongs the life of the process equipment. When the control point of the controller is at a high value, or when process load is large, a minimum flow setting may be required.

Single-speed floating controllers do not ordinarily have adjustable floating speed. If means such as an adjustable electrical interrupter are provided, the floating speed should be adjusted until the reduction of amplitude of cycle of the controlled variable is balanced against the increased period of the cycle. The period of the cycle should not be too long since slow floating speeds do not permit proper correction of load changes.

The controlled variable can generally be stabilized within the neutral only when process lags are almost negligible. The neutral usually does not need to be greater than 1 per cent of scale. If load changes are such that excessive deviation is caused, widening the neutral will not correct the situation.

For *proportional controllers*, whether the adjustments are continuous or in finite steps, it is only necessary to reduce the proportional band by increments until the desired stability is obtained. At this point it is desirable to check the adjustment by moving the control point a small distance from its previous setting, holding for a few seconds, and returning to the exact previous setting. This may be done only when process conditions permit such procedure by authorized personnel. If this disturbance causes a recovery similar to the kind desired, or similar to those previously shown in Fig. 8–10, the controller is properly adjusted.

For *proportional-speed floating controllers* the floating rate should be set originally to a low value (slow) and then gradually increased until the desired stability is obtained. The setting may be checked by using the same procedure given above.

Proportional-reset controllers require the adjustment of two control responses: proportional band and reset rate. Although many individual and valuable methods have been devised for adjusting controllers of this type, the following method is becoming a generally accepted procedure.[4]

1. Set the reset rate to zero or to as low a value as possible.
2. Determine the proportional band as outlined above.
3. Check the proportional band setting, and while doing this note the

period of cycling in minutes. (The period of cycling is equal to the time between two successive peaks as the controlled variable cycles on a recovery.)

4. Set the reset rate to 1 divided by the period of cycling in minutes. For example, if the period of cycling is 5 minutes, the reset rate should be set to 1 divided by 5 or 0.2 per minute.

5. Again check the stability of control on a recovery, and trim each adjustment as needed.

Many controllers of this type and those with rate response are not calibrated in terms of proportional band in per cent, reset rate in number per minute, and rate time in minutes. The manufacturer should be asked to supply these calibrations, or they may be determined by actual test.

FIG. 8–12. Effect of Control Responses on Period of Cycling.

Proportional-reset-rate controllers do not always have three separate adjustments, one each for proportional band, reset rate, and rate time. Two of these three adjustments may be synchronized so that the adjustment of only two is required. In some controllers a fixed rate time is incorporated so that it requires no adjustment. If the two adjustments are proportional band and reset rate, the procedure given above for proportional-reset controllers may be followed.

If three separate adjustments are included, the procedure may be:

1. Set the reset rate and rate time to zero or to as low a value as possible.

2. Determine the optimum proportional band, and while doing so note the period of cycling in minutes.

3. Set the rate time to one-eighth of the period of cycling in minutes.

4. Reduce the proportional band by one-fifth.

5. Check these settings, and note the new period of cycling in minutes.

6. Set the reset rate to 1 divided by the new period of cycling in minutes as found in step 5.

7. Check these settings, and trim if necessary.

It is sometimes difficult to determine which of the three responses are improperly adjusted when the controlled variable indicates excessive cycling. Figure 8–12 shows the influence of the three control responses

if they could be used separately on the same process so as to produce continuous cycling. Reset or proportional-speed floating response tends to produce a period of cycling approximately twice as long as proportional response. Rate response tends to produce a period of cycling about two-thirds that of proportional response.

When the process is of such a type that it is necessary to bring it on stream with approximately correct controller settings the adjustment is made doubly difficult. With such applications a careful analysis of the process reaction is a necessity. It is sometimes possible to refer to processes of similar nature, but such reference should only supplement process analysis, not replace it.

Systematic Reaction in Controlled System

The advantages gained by the application of automatic control to a process are partly due to action of the control system in responding to a definite change in an exact and consistent manner. Consequently, the stability and quality of control are based on a definite type of process reaction. As long as the process reaction is consistent, or systematic, the quality of control can be maintained.

The characteristics of the process, its capacity, resistance, and load, are not always fixed quantities, and in certain applications they may gradually change over a period of time. Gradual changes in process characteristics may mean a readjustment of the controller in order to maintain the quality of control. For example, gradual deposition of foreign material on the tubes of a heat exchanger or boiler increases the transfer lag. The proportional band of the controller may require widening in order to avoid excessive cycling.

Instability of process reaction may sometimes be encountered, more often in continuous chemical processes than in any other. Exothermic chemical reactions often display instability when temperatures are allowed to rise too far above the control point. The process reaction rate then increases to such magnitude that the temperature continues to rise even against controller corrective action.

Process dead time is frequently a changing characteristic since it depends upon fluid velocity and rates of flow to, through, and from a continuous process. Since no reaction is obtained until the dead-time period has elapsed, changes in its actual magnitude may have a profound effect upon controller adjustment.

Exact and consistent response of the measuring and controlling means to changes in the controlled variable is imperative. The almost universally applicable statement may be made that any unsystematic action of the control system results in a quality of control having greater

deviation and more continuous cycling than a carefully engineered system.

Erratic action of the control system caused by meaningless fluctuations in controlled variable, ambient temperature changes, and similar occurrences, requires that the controller make unnecessary corrective action since these actions carry through the entire control system. Such action may usually be corrected without excessive rearrangement by installing the necessary controllers for associated variables.

An appreciable dead zone in either measurement or control action may be the cause of difficulty in obtaining the desired quality of control. If the controller fails to respond to changes in the controlled variable, or if the final element fails to respond to changes dictated by the controller, the controlled variable must deviate further before the desired control action is obtained.

A dead zone in measurement results in undesirable error of indication of the controlled variable at the controller. Dead zone of measurement must be small, especially if the process has transfer lag or if the measuring lag is appreciable, in order to obtain control action with the slightest change in controlled variable.

Dead zones through the control system are, on the average, additive. Therefore when the controlled variable changes slowly, either because of large capacity or large transfer lag, the delay caused by the time necessary to traverse the width of the sum of the dead zones may become appreciable.

With floating or any type of proportional control a reduction in dead zone of measurement results in two advantages which are too important to be overlooked. First, the delay in corrective action is reduced. This reduction in dead zone permits a proportionately smaller proportional band or a faster floating rate. Second, the control response is made greater, and any dead zone in the final control element is itself made smaller, in terms of the controlled variable.

Inconsistent action of the final control element nearly always results in greater deviation and persistent cycling of the controlled variable. In the operation of valves, dampers, and louvers, a smooth characteristic is as important as the actual shape of the characteristic. A smooth characteristic allows small, gradual changes in flow to be made and results directly in stable action of the controlled variable.

Dead time in the control system, in addition to that caused by a dead zone, actually amounts to an unsystematic response since a delay or discontinuity results. Periodicity in either the measuring or controlling means introduces dead time which is detrimental in almost any control system.

Control of Batch Processes

A process in which the materials are treated in a vessel or furnace and the reaction completed as a single operation is termed a batch process. The time to treat each batch is usually not more than a few hours. This type of process is generally easily distinguished from the continuous process where the materials flow continuously and emerge as a treated product.
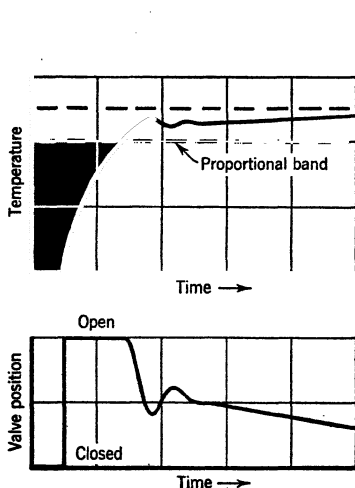
Fig. 8–13. Proportional Control of Batch Processes.
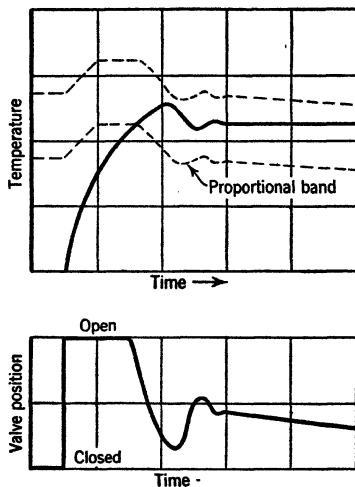
Fig. 8–14. Proportional-Reset Control of Batch Processes.

Batch processes are most often of the thermal type where materials are added to a vessel, the vessel is brought up to temperature, and the temperature is controlled for a period of time. The supply capacity is likely to be substantially smaller than the demand capacity. The process reaction rate is slow, and transfer lag and dead time are small. In such applications, a two-position (on-off) controller is generally satisfactory.

If the process reaction rate is large or if transfer lag is present, it may be necessary to employ proportional control in order to avoid excessive cycling of the temperature. Suppose that we consider the action of the controller through one complete batch. Figure 8–13 shows the temperature and the corresponding valve action with proportional control.

As the temperature rises when the processing begins, the valve is open. When the temperature reaches the lower edge of the proportional band

the valve begins to close. The temperature then cycles about the control point before becoming stable. With most batch processes, less heat is required as the contents of the vessel become more fully heated. Consequently, a gradually decreasing valve setting is required in order to balance heat losses.

With proportional control, the gradual closing of the valve can be accomplished only by a corresponding deviation of the variable from the control point. As long as the proportional band setting is small, offset will also be small. If the deviation should become large, the temperature may be maintained at the control point by manual reset.

Offset may be eliminated by means of reset response so that the heat supply is automatically balanced against losses. The action of the proportional-reset controller is shown in Fig. 8–14. The offset of temperature is eliminated, and the temperature is maintained close to the control point after it has stabilized. Notice that the initial overshoot of temperature when approaching the control point for the first time is larger than with only proportional control.

The large initial overshoot is generally due to the action of reset response. During the heating-up period the temperature is, of course, below the control point. The normal action of reset response is to shift the proportional band to urge the temperature toward the control point. Since the valve is already full open, reset response simply shifts the proportional band all the way above the control point. Consequently, the temperature must deviate considerably above the control point before the valve is closed enough to stop the temperature rise.

Here reset response produces a deviation greater by about one-half the proportional band than in proportional control, assuming that the heating period is sufficient to allow the reset to move the proportional band to its extreme. The deviation beyond the control point must encompass a large area in order to balance the previous deviation below the control point, as shown by the curves of Fig. 8–14. Since reset response is proportional to the area under the deviation curve, a considerable area above the control point must be obtained before the proportional band can be shifted to its proper position.

A controller with reset response is therefore of questionable value for the control of batch processes unless the initial overshoot is prevented by manual means. Adjusting the reset rate to zero during the heating-up period only, or rebalancing the reset action by momentarily setting a very high reset rate, are possible methods. Automatic limiting means on the reset response are provided in some types of mechanisms so that the overshoot of temperature on the initial approach to the control point is minimized.

Harmonic Cycling

In continuous processing, the output or product of one unit or plant may be the supply or feed to another unit or plant. Changes in the feed to a controlled unit or process constitute a load change. For example, if a chemical product being heated in a furnace comes from another process where it leaves at a controlled exit temperature, changes in its heat content produce a load change in the furnace.

A load change may occur in cyclic form, as in the example above, where the exit temperature of the previous process cycles as the result of the action of its control system. Nearly any controlled industrial process is a cyclic system having a definite period or frequency of cycling. Recovery curves generally show definite periods of cycling for different systems. This period is affected by process characteristics and controller adjustments.
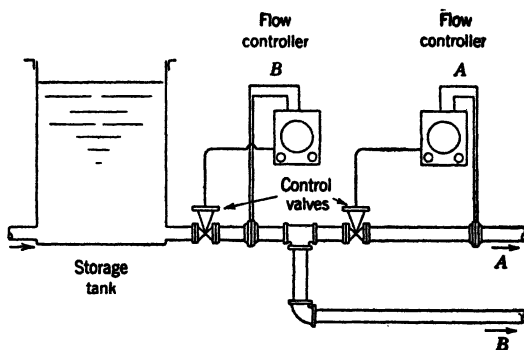
Fig. 8–15. Split Storage Control System.

A cyclic load change having a period and phase relation near or equal to the period and phase relation of a controlled process in which the load change occurs may cause harmonic action similar to the phenomena of sympathetic vibration in mechanics. If the phases of the two periods are in proper relation, the controlled variable in the process may cycle with large amplitude. An even worse situation is created where the second cycling condition reacts back into the changing load.

The two flow-control systems of Fig. 8–15 are examples of this type of arrangement. Let us suppose that the periods of each controlled system are alike. If flow controller A cycles in correcting for a deviation of flow, the flow through both lines A and B is caused to cycle. The cyclic flow through B may cause flow controller B to cycle, which

in turn reacts back to flow A. Thus harmonic cycling is set up in the complete system, and the actual control of the magnitude of each flow is lost.

Harmonic cycling can sometimes be avoided by changing the period of cycling of the controlled system so that one period is quite different from the other. Changing of the period can be accomplished either in the process or in the control system. Changing the process may be more difficult but is generally a permanent solution. Changing the control system is easier but may not offer complete freedom from harmonic cycling. In addition, a readjustment of the controller to avoid harmonic cycling usually sacrifices the quality of control.

The process can sometimes be changed to alter either the period or phase. The relocation of a valve or flow line, or the addition of capacity in the form of a surge tank, is often sufficient to correct the difficulty. The object is to change the process lag. Any method by which the process lag is either increased or reduced has the greatest influence upon the period of cycling.

The adjustments of the controller may sometimes be selected so that completely overdamped control action is obtained. Overdamped recovery is obtained by widening the proportional band and decreasing the reset rate on one of the controllers so as to increase the period of the system. Sometimes it may be necessary to increase the measuring lag or controller lag in order to obtain a different period and phase.

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CHAPTER 9

APPLICATION CONTROL ENGINEERING

The final test of any automatic control system is its performance during actual plant operation. The function of a controller is to utilize and complement the dynamic stability of the process. If the characteristics of each part of the controlled system are balanced and coordinated, excellent quality of control is obtained. A weak section at any point in the controlled system affects the complete system.

Many processes are comparatively simple and possess a certain degree of inherent stability. On the other hand, some processes possess a confusing array of capacities, lags, and load changes. Between these two extremes lie a large number of processes, complex to a varying degree, and requiring careful analysis.

As a preliminary step, this analysis should include a separation of the variables to be controlled. Each controlled process can then be studied individually from the standpoint of capacity, lags, and load changes. Until this analysis has been made, there is little likelihood that the process can be successfully controlled.

The control system should be properly engineered, special care being taken in the selection of primary and final elements. In too many applications the controller is handicapped by being required to overcome such factors as a large measuring lag, or a large dead zone in the control valve.

CONTROL OF FLUID FLOW

The control of rate of fluid flow not only is important in continuous-process industries but, as a relatively simple process, it represents one of the easier problems in automatic control. Flow control may well be termed the foundation of continuous processing because without it satisfactory automatic control of temperatures and pressures would be virtually impossible.

Fluid flow, for purposes of control, is usually measured by means of differential pressure. The mercury manometer flowmeter is in most general use, but the bellows-type flowmeter and the area flowmeter are also common. Few if any volumetric or current-type flowmeters are used for automatic control of flow, and the orifice plate is preferred

for its simplicity and ease of replacement. When the orifice is located at a distance from the controller, the measuring lag must be reduced to a minimum. Pneumatic transmission or the electric-type flowmeter may be employed for this purpose.

The control of flow in a pipe line can be accomplished by a control valve in the line when the system is under pressure supplied by hydrostatic head or by a centrifugal pump. This method, illustrated in Fig. 9–1, presents the possibility of placing the control valve either upstream or downstream of the primary element. A general rule is to arrange the control system so that the pressure most constant, either upstream or downstream, is one of the pressures at the orifice. It is essential that one of the differential pressures at the orifice be absolutely constant if it is desired to maintain a constant flow of gas.
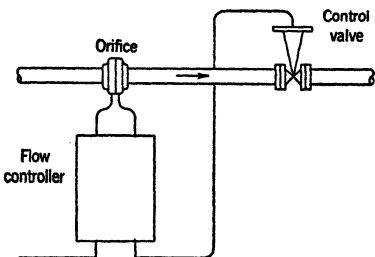
FIG. 9–1. Simple Flow Controller Installation.

If the control valve is installed in the discharge line of a positive-displacement pump, precautions must be taken against developing excessive pressure. An excess-pressure regulator in the line may be set to shut off the steam or power mechanism when excessive pressures are developed from an inadvertent closing of the control valve. An application of an excess-pressure regulator is illustrated in Fig. 9–2. The discharge of the pump in this illustration is controlled by returning a part to the suction side. Any increase in flow results in a greater opening of the control valve, more flow is bypassed, and the delivery is returned to the desired rate. The bypass around the control valve should remain closed in order to maintain high rangeability of the valve.

Probably the best method of controlling the discharge of a positive-displacement pump is to place the control valve in the steam line to the pump rather than directly in the discharge line, as shown in Fig. 9–3. A greater flow of steam, for example, results in greater pump speed and, consequently, greater delivery. Adjusting steam flow avoids large control valves in the delivery line when the quantity flow rate is large.

One of the most difficult examples of flow control is encountered in the speed control of a turbine-driven centrifugal pump. The arrangement is similar to Fig. 9–3 except that the pump is turbine driven. The control valve may be installed directly in the steam line, or a pneu-

matic diaphragm may be fitted to the turbine throttle valve, the hydraulic governor being interposed between the pneumatic diaphragm and the turbine throttle valve. The hydraulic governor adds stability to the positioning of the throttle valve, which usually has an extremely small travel.
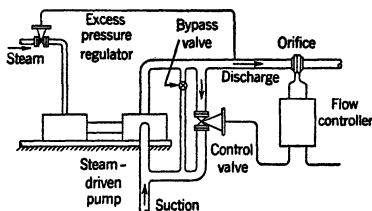
Fig. 9–2. Flow Controller Installations in Which the Discharge of a Positive Displacement Pump is Bypassed.
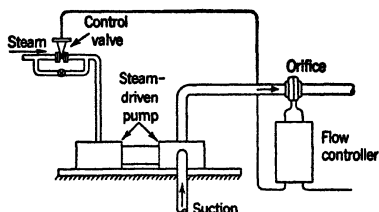
Fig. 9–3. Flow Controller Installations in Which the Steam Flow to Pump is Adjusted.

The reaction rate of flow to changes in valve position is almost instantaneous except when pump speed is controlled. Dead time due to the time required for the fluid to pass between the valve and orifice is usually quite small. Therefore, the only significant lags in the controlled system are the measuring and controller lags. The self-regulation of a flow process may be appreciable because of the head-flow relationship.

The proportional-reset mode with a pneumatic controller is best suited to processes of this kind, especially if load changes are present. Reset response not only provides corrections for load changes but also exerts a proportionate share of the control effort when lags are small. However, proportional control is feasible when load changes are small. For example, load changes are almost entirely absent when controlling the discharge of a centrifugal pump into a vessel held at constant pressure. Under such circumstances, offset is negligible even at a wide proportional band.

Satisfactory results can be obtained with a hydraulic controller having a proportional-speed floating mode, if all lags in the system are small, since stability is provided by process self-regulation. Obviously two-position control cannot be used because of the absence of any capacity in the system.

Where pump speed is controlled the characteristics of the pump enter into the controlled system. When inertia of the pump is small, the small process reaction rate due to the pump allows a small proportional band. However, in larger installations the inertia of the pump

causes transfer lag due to the gradual acceleration of a flywheel or turbine rotor, so that a wide proportional band may be required. In controlling pump speed, changes in steam pressure constitute supply changes in the system, and if excessive deviation of flow rate is to be avoided the steam pressure must also be controlled.

The control valve in any flow control application must be properly sized and carefully installed to maintain the required pressure differential. A semi-logarithmic flow characteristic is desirable only in order to maintain at least a linear flow characteristic if variations in pressure differential should be encountered. Otherwise, a linear valve characteristic may be satisfactory. A valve positioner is very often beneficial in obtaining accurate positioning and freedom from the effects of thrust and friction. Since the flow rate reacts almost instantaneously to changes in valve position, the action of the control valve must be smooth, steady, and exact.

Pressure Control

The control of pressure in vessels or pipe-line systems can unquestionably be classified as one of the simpler problems in automatic control, for two fundamental reasons: first, most pressure systems have relatively large capacity; and, second, very seldom is any appreciable transfer lag or dead time encountered.

The wide application of self-operated pressure controllers is made possible by favorable process characteristics. Controllers, often called regulators, for pressure service may be of the spring-loaded, weight-loaded, or pressure-loaded type, one of which is illustrated in Fig. 9-4. These devices have a small, fixed proportional band; consequently, the offset is small even for moderate changes in load. If smaller proportional band is desired, the diaphragm may be pilot operated by utilizing the high pressure of the fluid being controlled; this arrangement permits more accurate valve positioning.
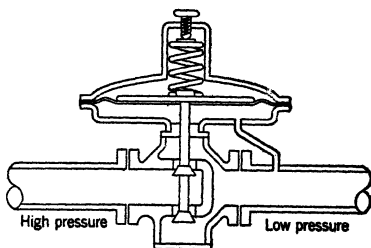
Fig. 9-4. Typical Spring-Loaded, Self-Operated Pressure Regulator.

When the pressure system has appreciable capacity and a slow reaction rate, two-position control may be used, especially if the valve is kept small so that the rate of change of pressure is small. Simple pressure controllers operating either a solenoid valve or a pneumatic diaphragm valve are common; they provide the advantage of remote

location of the valve from the point of measurement. All the simple pressure-control systems provide more satisfactory operation in the absence of rapidly changing loads and surges in supply pressure. The control valve characteristic is not so important as smooth changes in position.

A pressure gage of the bellows or spiral type permits greater accuracy of measurement and control. Even when the gage is placed some distance from the point of measurement, the measuring lag is small with gases and vapors. With heavy, viscous liquids the measuring lag is materially increased. The connecting line to the primary element should be neither too small nor of great length. Proportional control is always desirable.
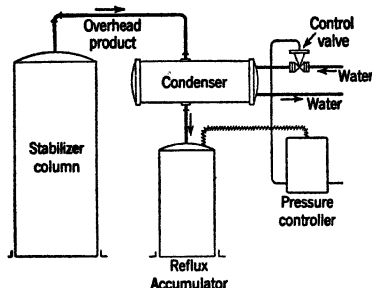
FIG. 9-5. Stabilizer Column Pressure-Controller Installation.

In the more complex pressure systems with several capacities, some transfer lag may be encountered. If load changes are present, the proportional-reset mode is required in order to eliminate offset. For example, the pressure in the stabilizer column of Fig. 9-5 is controlled by adjusting the flow of cooling water to the condenser. Gases from the column will be condensed if their temperature approaches that of the cooling water. Therefore, the rate of heat removal from the gases must be carefully controlled in order to hold a fixed pressure. A proportional-reset mode of control is necessary, and a large proportional band is indicated. Even then, control may be difficult when large load changes are present.

A simpler method of controlling column pressure is to control the release of non-condensable gases from the reflux accumulator. Transfer lag is almost entirely avoided, and a simple proportional controller is generally satisfactory. Better results are often obtained by slightly rearranging the process or the control system than by complicating it with supplementary controllers or control functions.

CONTROL OF FURNACE PRESSURE

In gas- or oil-fired furnaces it is often necessary to control the pressure within the furnace to promote efficient combustion. The pressure inside the furnace may be either slightly above or below atmospheric pressure, depending upon the type of furnace. Usually the pressure can be measured in terms of inches of water pressure — a full scale

range of 0.2 inch of water pressure being most common. A bell- or diaphragm-type pressure controller is used.

The point of measurement of furnace pressure must be carefully selected; it may vary considerably for different applications. Wide differences of pressure can often be found in the furnace, depending upon draft and flame conditions. The connecting line from the furnace to the controller must be large enough to avoid measuring lag. Compensation for changes in barometric pressure is nearly always required.

Furnace pressure is generally controlled by positioning a damper or a louver in the furnace draft duct. Control of fan speed is another method. The process has small capacity and negligible transfer lag, and the reaction of furnace pressure to changes in draft setting is almost immediate.

These process characteristics point to the choice of either a proportional-speed floating mode or a proportional-reset mode. The appreciable process self-regulation provides some stability with proportional-speed floating control. Reset response with the proportional mode is mandatory since wide variations in damper settings are always required in order to compensate for varying load and draft conditions.

Fig. 9-6. Typical Control Record for Furnace Pressure.

Pneumatic controllers or hydraulic control systems are common for the control of furnace pressure, the final element being positioned by a pneumatic or hydraulic cylinder. Usually the dampers or louvers are very large, and great power at the final element is required. Electric single-speed or multiple-speed floating control systems are also used with the damper positioned by an electric motor.

One difficulty often encountered is the proper sizing of the damper or louver and the proper adjustment of fan speed for forced draft. Without proper sizing, sudden changes in draft cannot be properly counteracted. If the draft system is oversize the controller is required to make minute corrections and the control system operates near the limit of its sensitivity. Louver and damper flow characteristics do not favor exact control, and advantage should be taken of linkage characterization of motion whenever possible.

A typical control record for rotary-cement-kiln pressure when proportional-reset control is applied is shown in Fig. 9–6. The pressure is held on the average to within 0.008 inch of water. What appears to be continuous cycling is actually variation in pressure within the kiln due to the combustion conditions with coal firing. Notice that the load change encountered at 1.6 hours on the record is quickly corrected and the pressure is returned to the control point.

Liquid-Level Control

Arrangement of the control system for the control of liquid level depends entirely upon the purpose which the control system is to accomplish. Where an exact level is to be maintained the control of the liquid level becomes very similar to pressure control. The ease with which liquid level can be controlled depends largely upon the capacity of the vessel being controlled. Transfer lags are generally small, although it is possible to have systems with appreciable dead time.

The two-position mode in either an electric or pneumatic controller is often acceptable in applications where only an approximate level has to be maintained. A great many direct-connected float-type level controllers are used for such services; since their float travel is generally about 6 to 8 inches for full valve travel, they can be considered proportional controllers within these limits. An electronic photocell relay may be applied if the level can be made visible to the photocell relay by means of a glass. A photocell relay controller is of the two-position type. A contact electrode, relay, and electrically operated valve are sometimes combined for two-position control of level when the electrode can be suspended over the surface of the liquid.

When wide variations are permissible, a two-position controller with a wide differential is satisfactory. As the level rises to the upper limit, either an inlet valve is closed or an outlet valve is opened. The reverse of this action occurs when the level drops to the lower limit. If the process capacity is large, the level will gradually and continuously move from one limit to another. Proper sizing of the control valve is necessary in these applications because it is easy to overflow or empty the tank if the valve is improperly sized.

If considerable static head is involved in control of liquid level, self-regulation may become important. If the vessel is not under pressure, the height of the level above either the discharge or inflow valves causes appreciable self-regulation. Overflowing or running dry is not so likely to occur, since the influx of liquid will decrease as the level increases, or the efflux of liquid increases as the level increases.

Automatic control is then applied only to keep the level within desired limits.

When the vessel is under pressure, self-regulation is generally negligible for the reason that the change in level constitutes only a small portion of the total head. For example, suppose that the vessel is under 100 lb per sq in. pressure and the level is 20 ft of water. A 10 per cent change in level constitutes only 0.8 per cent change in total head. If the vessel were not under pressure the change in total head would, of course, be 10 per cent.

FIG. 9–7. Liquid-Level Controller Installation.

There is a tendency in continuous chemical processes to reduce storage capacity and thereby reduce the initial cost of equipment. Consequently, the narrow proportional-band controller or the two-position controller is not often applicable for exact control of level. The absence of large process capacity combined with transfer lag due to multiple capacities necessitates proportional-reset control. Pneumatic controllers are nearly always selected.

In these applications the measuring lag should be kept to a minimum by proper installation of the liquid-level-measuring means. The differential-type level controller should be as carefully selected as the flowmeter counterpart. The float-type unit is often desirable for the measurement of liquid level since it is simpler and more easily maintained. If the liquid is viscous or tarry, any type of liquid-level meter is difficult to maintain at its best operating efficiency.

An application requiring exact control of liquid level is shown in Fig. 9–7. The storage of fluid in the tank is controlled by a float-type level controller with pneumatic proportional control. The outflow from the tank is usually controlled separately. The inflow is provided by two sources, one a fairly constant flow, and the other controlled by the level controller. Obviously, the system has minimum transfer lag and moderate capacity. A small proportional band is indicated, and the level can be held to quite close limits.

Averaging Liquid Level

In many applications, particularly in the process industries, the outflow of one unit becomes the feed to a succeeding unit. In order

to obtain stability of operation in the plant, it is important that fluctuation of outflow and inflow be reduced to a minimum, thereby maintaining all feeds relatively constant. The storage capacity of the vessel may be utilized to proportion outflow against changes of inflow. The vessel thereby serves as a surge tank for absorbing fluctuations in flow rates. Averaging control gets its name since the outflow is " averaged " against level, and the level is controlled between upper and lower limits rather than at a single point. However, the methods employed are quite different from those for controlling with a wide differential in two-position control.
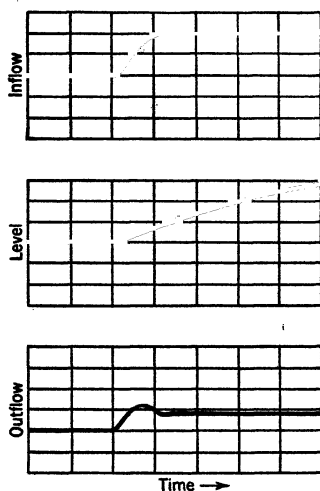
FIG. 9-8. Separate Control of Vessel Outflow.
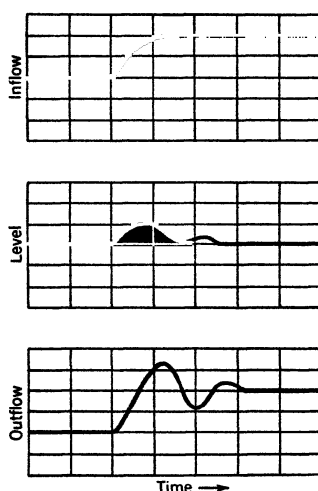
FIG. 9-9. Exact Control of Liquid Level by Adjusting Outflow.

Separately controlling the outflow of a single tank ordinarily results in wide variations in level. For example, Fig. 9-8 shows control of outflow with a proportional-reset controller and the resultant changes in level with changes in outflow. The outflow is maintained constant but only at the expense of large changes in level. Note that the gradual change in level constitutes a slow load change, and the outflow must deviate from the control point a constant amount. If the higher inflow should persist, the vessel will completely fill up.

Controlling the level of the tank as shown in Fig. 9-9 with the pro-

portional band set to the smallest stable value merely results in corresponding changes of outflow with inflow in order to hold the level at an exact value. It is obvious that if the vessel is to absorb fluctuations in inflow, a compromise between close control of level and separate control of outflow must be obtained.

Suppose that a level controller is set with a very wide proportional band, such as 300 or 400 per cent of full scale, and that the reset rate is maintained at a comparatively low or medium setting. As shown in Fig. 9–10, changes in level then produce only a small initial movement of the valve controlling outflow.

The level then drifts considerably from the control point, and the valve gradually moves to accommodate the new conditions. Reset response is relied upon to return the level gradually to an average point. Sustained changes in inflow ultimately cause a change in outflow, but the control action is considerably overdamped so that all changes in outflow are made slowly. The capacity of the vessel is utilized to absorb the surge.

FIG. 9–10. Averaging Control of Level When Vessel is Used as a Surge Tank.

Careful adjustment of proportional band is required in order to take advantage of full vessel capacity without producing excessive variations in either outflow or level. The level should be maintained as near the middle of the vessel as possible in order to take up surges in either direction.

A pneumatic type proportional or proportional-reset controller is used for averaging control because of its ability to provide large proportional band settings. A valve positioner is almost a necessity in averaging control because slow and smooth changes in valve position are required. The pressure differential across the valve should be maintained as nearly constant as possible so that the outflow is more directly proportional to valve setting. A secondary flow controller is often necessary in order to control outflow more closely.

Control of Thermal Processes

Although many of the more complete continuous processes are mixtures of multicapacity systems, many temperature-control applications have only one significant capacity. In general, all thermal processes have considerably larger capacities than flow processes, pressure processes, or even liquid-level processes. For this reason, the process reaction rate is relatively slow, and in simple processes the transfer lag and dead time are usually small. Heat-treating furnaces, pressure cookers, and electrically heated salt baths are examples of simple unit-capacity processes with small transfer lag and dead time.

The actual measurement of temperature and the determination of its exact meaning with relation to process operation are generally more difficult than the corresponding determinations for any of the other variables. For example, in metallurgical furnaces, the actual temperature of the product being heated is not measured directly. The measuring element is located in some part of the furnace in which the temperature is assumed to be most representative of the results desired. But temperatures vary considerably in different parts of even small furnaces, and the correct point of measurement can be determined only by a study of the relationship between these temperatures and quality of the final product.

The effect of temperature-measurement lag must be taken into account in all thermal processes. The dynamic error in measuring elements with large lag may be the cause of considerable difficulty. Small-gage thermocouples are advisable, but their effectiveness may be lost through heavy protecting wells. It is much better to have lighter wells and renew them regularly than to burden the controller with lags encountered in thermocouple wells of heavy mass.

The difference between large and small measuring lag in on-off temperature control is illustrated in Fig. 9–11, which shows the control of salt-bath temperature by means of a thermocouple and protecting well, and a radiation unit and target tube. One primary element controls while the other merely records the temperature. Controlling from the thermocouple, the pyrometer shows a cycle of 19°, while the radiation unit shows an actual temperature cycle of 40°.

When the bath temperature is controlled from the radiation unit the temperature cycle is reduced to 13° while the thermocouple shows only 5° cycling. Not only is the dynamic error greatly reduced by a smaller measuring lag but also the actual control of temperature is considerably improved. Note likewise that the period of the temperature cycle is appreciably smaller when the measuring lag is small.

In controlling furnace temperature and room temperature, the

measuring element is installed in heated air or other gases. The measuring lag of a temperature primary element in air is generally much greater than that of an element installed in liquid. It is necessary
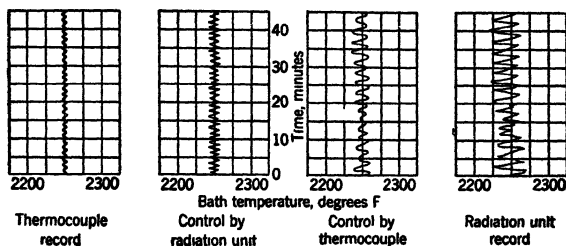
FIG. 9-11. Electric Two-Position Control of Salt-Bath Temperature (8-Gage Thermocouple in ¼-Inch Wall Chrome-Iron Tube; Radiation Unit with ¼-Inch Wall Ceramic Target Tube).

to avoid the location of the measuring element in a pocket or corner where the air is stagnant or moving with a low velocity, particularly if the measuring element receives no heat by radiation.

FIG. 9-12. Typical Control Record for Furnace Temperature Using Two-Position Control.

Most furnace and bath temperatures can be controlled with a thermocouple pyrometer and a two-position mode of control. Electrical furnaces are controlled by means of power relays which turn the electrical current to the heating electrodes on and off. Electric or electronic controllers with a solenoid valve or a two-position motor valve are common for gas-fired furnaces. Pneumatic on-off control is also suitable for fuel-fired furnaces. Figure 9-12 presents the control record for a gas-fired tool furnace with on-off control; it shows that, when transfer lag and dead time are small, a control cycle of 1 per cent is not difficult to obtain.

Single-speed floating control may be employed on processes of the same type with fast or moderate reaction rate. However, transfer lag and measuring lag must be an absolute minimum with single-speed floating control. When load changes are small, the temperature generally remains in the neutral, departing only occasionally. Single-speed

floating controllers of the electric type which interrupt the current to an electrically heated furnace at a constant rate but for variable fractions of a cycle are commonly used to provide single-speed floating control.

When the small cycle of temperature produced by two-position control or single-speed floating control is not desirable, proportional control with a narrow band may be chosen. Proportional rather than two-position control is often required for certain gas-fired heat-treating furnaces where controlled atmospheres are important. Load changes
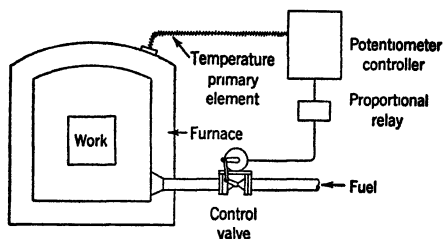
FIG. 9–13. Typical Furnace Temperature Control Installation Employing Electrical Proportional Control.

of moderate magnitude are allowable with only small resulting offset. The electric proportional controller is popular for applications such as that shown in Fig. 9–13. A potentiometer controller operates through a proportional or proportional-reset control relay to drive an electric-motor-operated valve. Furnace load is fairly constant, and fuel pressure is sometimes separately controlled with a self-operated regulator. Fuel-air ratio valves are also common in electric proportioning control. Where smooth and continuous valve action is necessary, the pneumatic proportional controller is advantageous.

Locating the measuring element nearer the source of supply of heat very often aids in the control of temperature. First, the time required for detecting change in temperature is reduced because there is less distance for the heat to travel; second, the variations of temperature are much greater nearer the supply, particularly in thermal processes. Locating the measuring element in direct view of the heating element or flame takes advantage of radiation and improves the detection of temperature changes.

In a heat exchanger, for example, it is sometimes possible to locate the measuring element in the demand side but in a position close to the supply side where the heating or cooling medium enters. More rapid indication of temperature changes is thereby obtained. Lo-

cating the measuring element near the supply may bring two disadvantages. First, the measured temperature is not directly indicative of either supply or demand temperatures but is somewhere between the two. Second, fouling of a heat exchanger may change the magnitude of the demand temperature without any indication of its changing. The advantages for control generally outweigh these disadvantages.

FIG. 9-14. Plating-Bath Controller Installation in Which the Supply Temperature is Measured and Controlled.

When process load is constant temperature may be controlled from either the supply or demand. An example is found in temperature control of plating tanks, illustrated in Fig. 9-14. Control is accomplished by supplying either steam or cold water to the circulating water flowing through the jacket around the bath. Ordinarily the temperature of the bath would be measured by placing the primary element directly in the bath. The large supply capacity, represented by the thermal capacity of the circulating water, produces transfer lag which makes the control of bath temperature difficult.

FIG. 9-15. Typical Record for Control of Plating-Bath Temperature.

The stability of control when controlling from the bath is poor, as illustrated in Fig. 9-15. The control record indicates that it is difficult to stabilize the temperature. When the primary element is placed in the circulating water, however, the bath temperature is immediately stabilized with relatively small deviation. The circulating pump maintains a high rate of heat transfer, thereby obtaining a minimum temperature difference between circulating water and bath.

Large potential difference between supply and demand increases the difficulties of temperature control because of the resulting higher rate of heat transfer. This statement is best illustrated by the oil heating furnaces in refinery service. Heat is transferred by radiation

from a source at 2000° F to oil at 800° F moving at high velocity in tube banks. This large temperature difference alone is sufficient to make the use of two-position control impossible.

Thus, the control of countercurrent heat exchangers where the average temperature difference is small is generally less difficult than the control of concurrent heat exchangers where the average temperature difference is large. In milk pasteurizers, for example, the temperature of the supply (warm water) is only a few degrees greater than that of the milk. This small temperature difference results in close control, and the possibility of even momentary deviations is reduced.

Control with Transfer Lag or Dead Time

Processes with a number of separated capacities usually possess sufficient transfer lag to prohibit the use of two-position control because of excessive amplitude of cycling. If the process also has a slow reaction rate, the period of cycling is likely to be excessive. In such applications proportional control is the only mode providing the proper amount and rate of corrective action. If load changes occur even to a small degree, reset response is necessary because of the wide proportional-band settings that are required.

Fig. 9-16. Fractionating Column Temperature Controller Installation.

The control of temperature in large continuous annealing furnaces, for example, must be exact, and load changes of varying magnitudes and rates cause constantly changing fuel requirements. Large side-capacities often contribute enough transfer lag to affect the operation of the controller. Electric proportional-reset control is used when the process reaction rate is slow, and a pneumatic proportional-reset controller when it is fast.

Heat exchangers with multiple capacities sometimes possess very large transfer lag. Such systems are difficult to control because of the wide proportional band required. An example of a multiple-capacity heat exchanger is the fractionating column shown in Fig. 9-16, in which the feed enters at the side of the tower and passes up through a large number of bubble trays. Each tray constitutes a thermal capacity and is separated from the succeeding tray by the resistance to flow of heat.

In most fractionating columns the overhead temperature range is

maintained constant. It is evident that in this example the stabilizing effect of capacity is outweighed by the transfer lag which it creates. The only way to obtain satisfactory control is to reduce the rates of load changes as much as possible by controlling separately nearly all the variables entering into the system. Temperature control of fractionation must always be accompanied by pressure control, of course.

In the control of thermal processes, especially those with transfer lag, sensitivity of measurement is of utmost importance. A dead zone of measurement converts transfer lag to dead time and adds to the dead time already present. With greater sensitivity of measurement each small change in temperature is detected, and a corrective action can be made by the controller. A measuring means with an appreciable dead zone allows a large initial deviation when a load change occurs. A small dead zone results in greatly improved control of temperature.

When dead time exists to any degree anywhere in the controlled system, a wider proportional band and slower reset rate are always necessary in order to avoid cycling of the controlled variable. Any load changes which occur in the dead-time period are unavoidable since the controller cannot provide corrective action.

When transfer lag or dead time is excessive, minimizing process load changes is preferable to attempting to control entirely from a single end temperature. It may be assumed that a continuous process may have more or less continuous changes in demand, and it is the purpose of the controller to counteract the effect of these changes. But to burden the controller with needless changes on the supply side simply results in excessive deviation which the controller cannot effectively prevent.

Therefore in applications of this kind very careful consideration must be given to eliminating any changes in supply. Placing a separate controller at the supply or introducing a secondary controller into the control system always results in marked improvement in the quality of control.

In proportional-reset control of continuous processes, rate of load change is as important as magnitude of load change. A load changing very rapidly or suddenly results in appreciable deviation with practically any control system. Conversely, a load changing at a slow rate may be counteracted with negligible deviation. Consequently, if all the variables affecting the operation of the process can be maintained fairly steady, the deviation of the main controlled variable can be held to quite close limits.

Rate response is most useful for counteracting the effect of dead

time. The advance of control response in terms of minutes rate time is often the only means of overcoming the delay caused by dead time. Temperature-control applications usually possess sufficient capacity to level out any fluctuations in valve position. Rate response may often be found useful for counteracting sudden changes in load which cannot be eliminated in any other manner. Since rate response reacts to the rate of change of the temperature, any sudden large change in temperature will cause a correspondingly large corrective action and deviation may be arrested before it can become excessive.

The valve characteristic for such applications is generally dependent upon the magnitude of load changes. If load changes are small, a linear valve characteristic may prove satisfactory. A semi-logarithmic characteristic is generally more desirable, however, if load changes are large. Rangeability of the final control element is important in temperature control because of frequent wide changes in load. As in other difficult control applications, a valve positioner greatly improves the operation of the control system.

Humidity and Air Conditioning

In industries like food, textile, and paper processing, the condition of the air is one of the most important influences upon the quality of the product. Moistening and drying operations are essential in the production of such materials as woven cloth and processed foods.

Air conditioning includes the control of any of the physical qualities of air. The functions involved in air conditioning include temperature, humidity, cleaning, and distribution. Automatic control of temperature and humidity is the basis for all conditioning systems.

The daily and seasonal changes in atmospheric conditions constitute the uncontrolled variables in the system. Actually four control functions are required in conditioning the temperature and humidity of air: heating, humidifying, cooling, and drying.

Although not all systems involve control of these four functions, large industrial applications generally employ complete control. Domestic air conditioning generally includes only one or two of these functions in any one application.

The heating and humidifying of the air are usually accomplished separately. The heating unit is controlled from dry-bulb temperature, and the humidifying unit is controlled from either wet-bulb temperature or relative humidity. The system is generally arranged so that the following cycle is accomplished:

1. Mixing of fresh with recirculated air.
2. Preheating.

3. Humidification.
4. Reheating.

The cooling and drying of the air may be accomplished by the same piece of equipment, since any cooling operation is automatically accompanied by dehumidifying. However, reheating is advisable if both temperature and humidity must be accurately controlled at only slightly reduced levels. The cycle is then represented by:

1. Cooling with attendant drying.
2. Reheating.

Combinations of these two cycles must be used in conditioning systems operating under varying load and atmospheric conditions. The condition of the supply air is the determining factor in arranging the cycle. Often a very simple cycle is satisfactory.

FIG. 9-17. Air-Conditioning Control System.

When dry-bulb temperature is controlled, the humidity may be measured either by a wet-bulb thermometer or by a hygrometer. The primary element, whether a thermometer bulb or a hair hygrometer unit, must be carefully installed so as to obtain not only accurate indication but also small lag. Whenever accurate measurement and exact control of humidity are required in industrial applications, the dry bulb-wet bulb method of measuring and controlling humidity is imperative.

Control of humidity generally involves large capacity as well as slight transfer lags. Two-position or proportional control is adequate if load changes are not severe. If capacity is small or load changes frequent, proportional-reset control is the proper method, especially where exact control is required, as in small testing chambers.

An application requiring controlled humidity and temperature is the processing of smoked meats. The control system is shown in Fig. 9-17. The fresh air, filled with incomplete products of combustion, is recirculated through the chamber at controlled temperature and humidity. The fresh air mixing valve and the humidifier are adjusted by the wet-bulb temperature. The heating coil is adjusted by the dry-bulb temperature. The primary elements are located in the return line from the chamber.

MANUAL CONTROL

Continuous processes may be started and brought to a preliminary balance by means of manual control. When the preliminary balance has been obtained the control systems maintain the balance of the process. Manual control is desirable because many of the processes in the unit must arrive at balance in a certain time relationship—some must be completely balanced before others, and some must balance simultaneously.
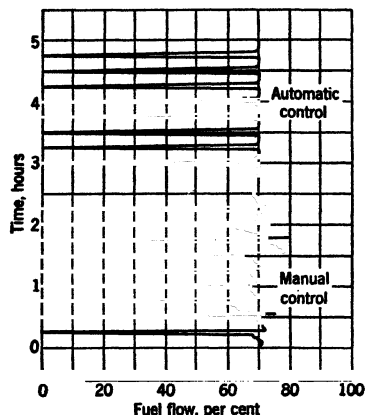
FIG. 9-18. Automatic versus Manual Control of Fuel Rate to Heating Furnace.

Thus, when a unit is brought on stream another variable in the form of t me is introduced and the ordinary control system has no provisions for automatically controlling the variable with time. Furthermore, arriving at a preliminary balance constitutes a major load change in the controlled system. It is difficult to adjust the control system to maintain a proper balance for such extreme changes.

There may be occasions when load changes may be so violent that the control system cannot properly counteract the effects. A large and sudden change may sometimes bring all the associated control systems into harmonic cycling. Manual control must then be resorted to in order to reestablish balanced conditions.

Although it is possible to select instances where expenditure for automatic control is not justified and where manual control is therefore necessary, there are few in which automatic control would not result in greater economy of operation or better quality of product. Manual control of a process can be made effective only when:

1. The process has inherent stability and moderate reaction rate.
2. There is no dead time.
3. There are few load changes.
4. Accurate control is not necessary.

Manual control must, however, be conscientiously applied by an operator in order to be effective, since the need for continuous correction is often overlooked. On the other hand, an operator can adjust the system on the basis of both past and future action without restrict-

ing the control to the reaction of one variable in the process. Rational guesswork is often effective in manual control.

If many of the conditions outlined above are not present, then automatic control is desirable because the response to the corrective action may be so immediate that continual correction is necessary. On the other hand, the response may be so very slow that overcorrecting would result from not being able to observe the effect of a correction reasonably soon. Minute and exact corrections can be made more easily by an automatic controller than by manual means.

In Fig. 9–18 are shown the comparative results of manual and automatic control of rate of fuel flow to an open-hearth furnace. The periodic changes in flow are due to the reversal of the firing arrangement in this application. Notice that the rate of fuel flow has considerable tendency to drift with manual control whereas automatic control holds an exact balance between reversals.

It must be remembered that a controller cannot think or anticipate but must base its action on events of the present. The control system is but an automatic machine for performing a tedious task which a human operator would be incapable of performing.

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CHAPTER 10

AUTOMATIC CONTROL SYSTEMS

Most control systems include only a single measured and controlled variable. It is obvious that in many applications the introduction of other variables into the control system is necessary for the more complete accomplishment of the purpose of automatic control. The control system must then be made responsive to changes in more than one variable.

When the controller is forced to counteract the effect of changes in two, three, or even more uncontrolled variables in the process, the quality of control is dependent upon the rate and magnitude of changes in these variables. Operation can be improved by arranging the control system so that several variables are controlled.

Pressure changes in the control agent are often the cause of difficulty, and *metered control* is used to introduce either pressure or flow as a second measured variable in the control system. Often the balance of the process or the operation under control is dependent upon a proportion or relationship between two variables. When this requirement exists, the control system must be arranged so that one of the variables is controlled in a fixed relationship to the other. In some instances the second variable introduced into the control system may be time; that is, a measured variable must be controlled to a certain magnitude with time. Proportioning of variables in this manner is accomplished in *ratio control* and *time-variable control.*

It is also necessary in many processes to protect either the process equipment or control system from the effects of excessive temperature, pressure, or other variable. An auxiliary controller is often connected into the control system to limit its operation so that excessive temperatures or pressures may be avoided. The control of sudden reactions in the process is often accomplished in a similar manner. *Limit control* and *series-proportional control* are methods by which auxiliary variables are introduced.

It is possible to indicate only the general principles involved in multivariable control systems since these systems are nearly always arranged to meet the requirements of a specific application. By ingenuity in composing standard arrangements and by the tailoring of special control

mechanisms to individual needs, many variables may be correlated into a single, extensive control system.

METERED CONTROL

Many processes are controlled by regulating the flow of a heating medium such as steam, fuel gas, or fuel oil for supplying heat to a process. Variations in flow of the control agent not dictated by the controller cause supply changes. Such variations in flow are caused by changes in pressure differential at the valve, which, in turn, result from changes in pressure of the fuel supply, and changes in downstream pressure caused by clogging of fuel burners, and so on. These changes are difficult to counteract since they must carry through the process before they are detected by the controller. Supply changes sometimes occur suddenly or over a wide range, and deviation may become excessive before a new balance of conditions can be established.

Supply changes must be corrected before they enter the process if adequate control is to be maintained, especially where the process has large transfer lag or dead time. For example, in the heating furnace of Fig. 10-1, the rate of flow of fuel oil is determined by a controller which measures a temperature in the furnace. Ordinarily the control valve would be set directly by the controller. Suppose, however, that a flow controller is installed in the oil line and its control point automatically regulated by the master temperature controller. With pneumatic controllers, for example, the output pressure of the master controller is connected to a receiving mechanism in the flow controller. The control point is made to move over the scale of the flow controller in response to the output pressure of the master controller. We now have a system in which the master controller can call for any rate of flow within the limits of the flow controller. This flow is maintained at the desired value by the separate control action of the flow controller.

FIG. 10-1. Metered Control System Employing a Secondary Flow Controller.

The flow controller insures a constant rate of flow of control agent regardless of changes in pressure differential across the control valve. Variations in oil pressure cause a change in flow which is immediately corrected by the flow controller without any action by the master con-

troller and without upsetting the control of temperature. At the same time, the master controller adjusts the valve position in order to control the magnitude of the master variable. This control action is accomplished through the flow-controller mechanism.

FIG 10-2 Metered Control System Employing a Secondary Flow Controller for Averaging Liquid-Level Control.

The above-described system of metered control is extensively applied in averaging control of liquid level as illustrated in Fig. 10-2. A liquid-level controller regulates the control point of a flow controller. Pressure changes in the vessel and downstream from the control valve frequently make the control of liquid level in the tank difficult. By means of metered control the effect of these pressure changes can be eliminated. Both liquid level and outflow can thereby be made more stable and consistent

A system of metered control which accomplishes similar results is shown in Fig. 10-3. Instead of a flow controller, a pressure controller

FIG 10-3 Metered Control System Employing a Secondary Pressure Controller
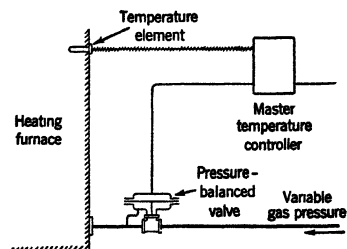
FIG 10-4 Metered Control System Employing a Pressure-Balanced Control Valve

regulates the pressure of the control agent. This system is useful only when the control agent is a fuel gas or steam. If the hand valves or other orifices at the burner remain in fixed positions, the pressure is directly proportional to flow. The pressure controller serves to change the gas or steam input by varying the downstream pressure of the

control valve. The action of the pressure controller is, of course, identical to the action of a flow controller.

A third method of accomplishing metered control is shown in Fig. 10–4. A pressure-balanced valve corrects for variations in gas pressure. The valve is so constructed that the output pressure of the controller is applied to one side of the diaphragm and the downstream gas pressure is applied to the other side of the diaphragm. The pressure-balanced valve is actually a self-operated pressure regulator whose control point is regulated by the master controller. Any change in gas pressure is counteracted by the pressure-balanced valve and is not allowed to pass into the process.

The master controller must use whatever mode of control is required by process characteristics. Nearly always, however, this is the proportional-reset mode since metered control is applicable where load changes of appreciable magnitude are involved. A flow controller with regulated control point adopts the mode of control best suited to the control of flow: sometimes proportional, but more often proportional-reset, control. A pressure controller with regulated control point is more often a proportional controller.

Metered control does not maintain any fixed relation between the master variable and the control agent which is manipulated by the master controller. The flow or pressure is set to any magnitude required by the master controller in order to maintain control of the process. In this manner, the flow of the control agent is related through the process to the master variable. Actually the secondary flow or pressure controller is equivalent to a control valve in an ordinary control system.

In metered control it is necessary to consider characteristics such as controller scale shapes and valve characteristics. The problem involved is similar to that of the control system analyzed in a previous chapter where valve characteristics were related to magnitude of process loads. In metered control the problem is more complicated because of the greater number of factors.

Supply changes are corrected by the secondary controller; demand changes, by the master controller. For a load change caused by supply, only the valve characteristic is important since the magnitudes of both master and secondary variables remain essentially unchanged. In metered control with a flow controller, considering supply changes only, a linear or semi-logarithmic valve characteristic is satisfactory.

For a load change caused by demand, the characteristic of the secondary controller scale is important. The magnitude of changes in flow is governed by both the characteristic of the secondary controller scale and the characteristic of the mechanism which adjusts the secondary

control point. This mechanism usually has a linear relation between master variable and setting of secondary control point in inches of secondary controller scale. The secondary controller scale may be either linear or square root, depending upon the type of flowmeter. A square root scale, however, is not greatly different from linear between 40 or 50 and 100 per cent of scale. An approximately linear relation between flow and master variable is satisfactory as long as process load does not change over an extremely wide range.

Ratio Control

In many applications, especially in the control of continuous processes, it is desirable to ratio or proportion one variable with another. For example, the flow of two fluids must often be proportioned in order to maintain a suitable mixture.

Ratio-flow control systems are arranged as shown in Fig. 10-5. The primary instrument here is not a controller but a transmitter. The control point of the secondary controller is set in direct relation to the magnitude of the primary flow. For that reason it is important that the primary transmitter have a linear relationship between its measured variable and transmitted pressure or other output impulse.

Fig. 10-5. Ratio Control System for Two Flows.

As the magnitude of the primary variable changes, the control point of the secondary controller is automatically moved to a new value so that an exact ratio is maintained between primary and secondary variables. The secondary controller controls the magnitude of the secondary variable to the value called for by its control point.

The mechanism which adjusts the control point of the secondary controller should include zero and range adjustments, permitting variations in ratios and magnitudes of the two variables. Basic ratios are established by the ranges of the primary transmitter and secondary controller. The basic ratio may be selected by proper sizing of the orifices. For example, if each flowmeter has the same range, and the ratio of primary to secondary is to be 0.5, then the control-point range must be 200 per cent. In other words, the control point of the secondary controller must move over full scale for only 50 per cent of full-scale change of the primary variable.

The primary transmitter and the secondary controller must have identical scale characteristics; otherwise an exact ratio cannot be main-

tained. For example, a square root scale on the primary transmitter would not match a linear scale on the secondary controller. For that reason both primary transmitter and secondary controller must use either the square root or linear scale. In order to maintain the desired ratio accurately, the secondary controller must have proportional-reset control. Reset response is necessary in order to maintain the secondary variable at the shifting control point as closely as possible.

Ratio-flow control systems may also be employed to proportion several secondary variables to one primary variable. For example, it is possible to set the magnitude of two or more secondary flows in relation to a single primary flow by connecting the primary transmitter to the mechanisms for adjusting the control points in two or more secondary controllers.
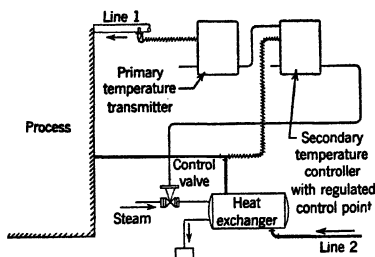
FIG. 10–6. Ratio Control System for Two Temperatures.

Ratio-flow control is also accomplished by other means, one of which is to place volumetric meters in both the primary and secondary flow lines. A differential-gear mechanism connects the two meters, and a control valve in the secondary line is positioned by the difference in rotational speeds of the two meters. Proportioning pumps are also arranged so that the ratio of two flows depends upon the speed and capacity of each pump.

Ratio control may be applied for variables other than flow. For example, it may be necessary to maintain a ratio of heat flows as shown in Fig. 10–6, such that the total heat flow in both feed lines remains constant. Assuming that the rate of flow in the feed lines is constant, the temperature of the feed in line 2 is maintained at a ratio of the temperature of the feed in line 1 by controlling the heater in line 2.

When the temperature of the feed in line 1 increases, the primary temperature transmitter lowers the control point of the secondary temperature controller. This, in turn, partly closes the steam valve to the heater in line 2 and lowers the temperature in line 2. The heat balance of the process is thereby maintained more exactly.

Ratio control also has applications where a controlled relation exists between temperature and pressure. In the handling of vapors, hydrocarbons for example, pressure as well as temperature determines its state. A specified state may be obtained by measuring the temperature of the vapor (primary variable) and maintaining pressure (secondary variable) at a specified ratio.

The important factor of process reaction rate should not be ignored in ratio control or in any other correlated systems. Suppose, for example, that we wish to maintain temperature as a ratio of pressure in a ratio-control system. Pressure is measured and used to determine the magnitude of the temperature. However, as pressure changes ordinarily occur at a fast rate, the control point of the temperature controller would be moving at a relatively fast rate. If the temperature process has a slow reaction rate, as it usually does, the maximum rate of change of temperature might be so small that the temperature could not follow the moving control point closely enough to maintain the desired ratio. In ratio-control systems it is obvious that the secondary variable must have a reaction rate equal to or greater than the primary variable; otherwise the ratio cannot be accurately maintained.

It should be noted that ratio control is altogether different from metered control even though the control systems have similar outward appearances. In ratio control a definite ratio exists between the primary and secondary variables at all times, and there is no reaction of the magnitude of the secondary variable back to the magnitude of the primary variable. In metered control there is no fixed relation between magnitudes of master and secondary variables, and the secondary variable acts through a process to control the magnitude of the master variable. In some applications, the primary variable may be controlled by the addition of a control system to the primary transmitter. This arrangement does not alter the operation of the system.

A pneumatic ratio-control system is perhaps the most common and the most versatile in operation. However, the primary and secondary variables can be combined in one controller and the control point operated mechanically. The principle of operation is the same in this type of system as in the pneumatic system. Electric and hydraulic-type ratio-control systems are also used, however.

Time-Variable Control

In the processing of metals and chemicals it is sometimes necessary to vary the controlled variable over a definite time schedule. For example, in annealing steel a schedule as shown in Fig. 10-7 may be required. The furnace temperature is raised to 1500° F in 4 hours, held for 8 hours, and lowered to 500° F in 6 hours. This time schedule must be incorporated into the automatic control system.

Time-variable control is accomplished similarly to ratio control. All systems operate on the principle that the control point is moved through the desired time schedule. The control system functions to maintain the controlled variable, temperature in the example above, as close to the moving control point as required.

There are four principal methods for obtaining time-variable control. In the first arrangement an electric motor drive is attached to the control point of the controller in such a manner that applying electric power to the motor moves the control-point index along the controller scale. The speed of the motor drive is selected to fit the particular application. Auxiliary electric timers and interrupters govern the movement of the control point in accordance with the desired schedule.
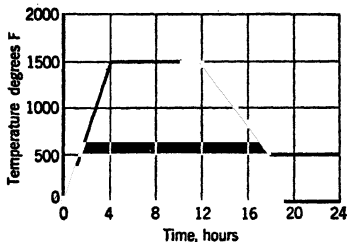
FIG. 10–7. Typical Temperature Schedule for Time-Variable Control.

A second arrangement employs a two-record self-balancing potentiometer. One record is operated in the ordinary manner from the controlled temperature which it is desired to maintain on a schedule. The second record is operated along the desired time schedule rather than being used to record a measured variable. The second record or point is positioned from a variable-voltage source in which the voltage is made proportional to the schedule by means of a shaped cam. By a simple arrangement, the second record is made to operate as the control point of the controller. The controlled variable (the first record) is then controlled to the magnitude dictated by the control point (the second record).

A third arrangement involves a transmitting mechanism whose output pressure is made proportional to the desired schedule. A cam in the transmitting mechanism is shaped to provide the desired time schedule. The transmitting mechanism adjusts the control point of a secondary controller having a control-point setting mechanism. The secondary controller measures and controls the variable to the desired magnitude.

A fourth arrangement provides mechanical means for transmitting the desired schedule to the secondary controller and operating the control point. A cam shaped to the desired time schedule may be located either outside or inside the controller case. In this class we may place all the many simple types of time-temperature and time-pressure controllers of the non-indicating or indicating types found in the control of batch-type operations.

Although there are a multitude of time-variable control mechanisms and systems, the application of one type is illustrated in Fig. 10–8 for time-temperature control of a crystallizing kettle. The time-schedule mechanism contains a cam shaped to provide a schedule as follows: uniform heating rate to 212° F during 2 hours, a holding period of 2

hours at this temperature, and a cooling period with a uniform temperature decrease of 18° F per hour for 3 hours, and then a uniform decrease of 7.2° F per hour for 13 hours. The complete cycle takes 20 hours.

Control of temperature during the time schedule is accomplished with the proportional-reset mode of control by regulating steam to the kettle jacket. It is in applications of this general nature that a large number of simple, non-indicating time-temperature controllers are used.

As in ratio control any number of variables may be connected with the same time schedule. In multiple-zone furnaces, for example, the time-variable control system is usually arranged so that each zone is separately controlled, but all to the same time schedule.

Fig. 10–8. Time-Variable Control System for Kettle Temperature.

In all these systems the controller must maintain the variable at a moving control point much in the same manner as in ratio control. Consequently it is necessary to employ a mode of control which will maintain the controlled variable at the control point within the desired limits. The selection of the mode of control is based upon the process characteristics and the desired quality of control, just as with any control system.

Two-position and single-speed floating control find wide usage in time-variable control of thermal processes where transfer lag or dead time is negligible. Proportional control is suitable only if the proportional band may be made small, thus eliminating any serious offset. The proportional-reset mode is ordinarily best adapted to time-variable control since reset response tends to eliminate offset caused by varying positions of the control point.

Process reaction rate again must be considered if a fast rate of change of control point is required. The control point should not be moved at a rate greater than that at which the controlled variable can change; otherwise deviation of the variable is inevitable. Ordinarily the rate of control-point movement is quite slow as illustrated by the example in Fig. 10–7.

Careful consideration must be given to the final element because of the wide variations in flow required at various positions of the control point. Adequate sensitivity of valve positioning and smooth changes in valve position are necessary if close control is desired. The rangeability of the

control valve should be large to provide adequate control of both large and small magnitudes of the controlled variable.

LIMIT CONTROL

Many simple controllers serve in industrial process control as safety devices to protect process equipment from overloads of temperature or pressure. The addition of a second controller to the control system results in a simple type of multivariable control system.
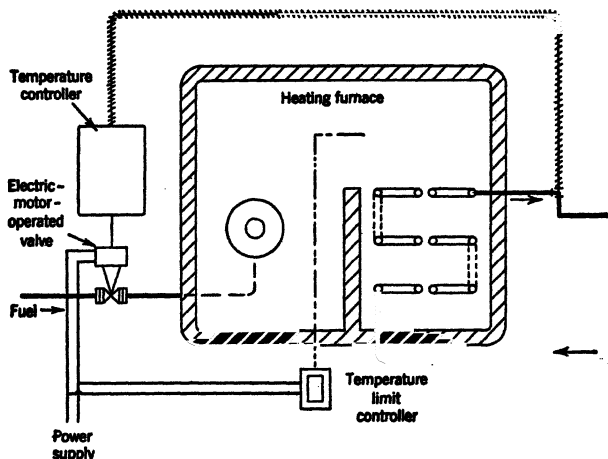
FIG. 10–9. Limit Control System for Preventing Excessive Brickwork Temperatures.

Figure 10–9 illustrates the application of a limit control on a gas-fired heater. At times, excessive gas supply may cause the temperature of the brickwork to rise above the softening point. A second electric two-position controller of the millivoltmeter type is installed with the thermocouple in the brickwork. The control contacts of the limit controller are connected into the control system so that the control valve ordinarily operated by the temperature controller is closed by excessive limit temperatures.

The main controller operates in normal fashion as long as the limit variable remains below the desired point. If the limit variable rises above the safe point, the control of the main variable is automatically cut off until the limit variable returns to the safe level. Note that,

during this period, control of the main variable is lost, and deviation of the main controlled variable may occur.

A "lockout"-type limit-control system is shown in Fig. 10-10. Temperature of the heater is controlled by a pneumatic controller and diaphragm valve. At times, however, the feed to the heater may drop below a safe limit and it is necessary to shut off the entire system until the flow of feed can be re-established. In the present example a flowmeter having electric limit contacts is installed in the feed line. When the flow of feed drops below the set limit, the contact is broken, the electrical locking-type relay is deenergized, and a three-way valve in the temperature controller valve line is opened. The fuel flow to the heater is cut off and a signal given. The system is then completely shut down until it is manually started.
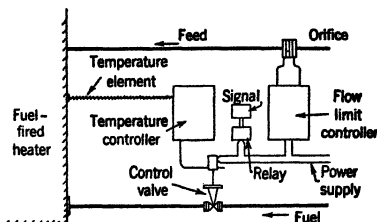
FIG. 10-10. Limit Control System for Closing down Process upon Feed Failure.

When the control system is electrically operated, a limit controller of the electric type is used to remove all power if the control valve is solenoid operated, or to drive a motor-operated valve to the closed position. With pneumatic controllers the limit controller is arranged either to remove all pressure or apply full pressure to the diaphragm control valve.

With fuel-fired furnaces, it is sometimes necessary to connect flame-protecting devices into the control system in order to protect the furnace from dangerous building up of unburned fuel if the flame should go out. The flame device is usually arranged so that the control system is shut down when the flame is out. These installations, in the interest of safety, must be made in an approved manner.

A series-proportional control system is appropriate when the process arrangement requires that the control be accomplished from either of two variables at different times. This type of multivariable system usually, although not necessarily, involves considerations of safety. Series-proportional control may therefore serve a different purpose from limit control.

Let us suppose that the reaction in the chamber of Fig. 10-11 is controlled with a proportional temperature controller. At a certain time the reaction in the chamber causes the pressure to rise rapidly above its normal value. Assume also that during this period pressure is more indicative of process conditions than temperature. For this type of

process a pneumatic proportional controller to control pressure is installed in series with a temperature controller, both operating a single diaphragm control valve.

The action of the system normally is such that the temperature controller maintains a moderate output pressure, 8 lb per sq in., for example. *The measured pressure is normally such that it is below its control point,* and the pressure controller calls for a high output pressure, 12 lb per sq in. or more. However, since the output of the pressure controller becomes the supply to the temperature controller, only 8 lb per sq in. is maintained at the valve.
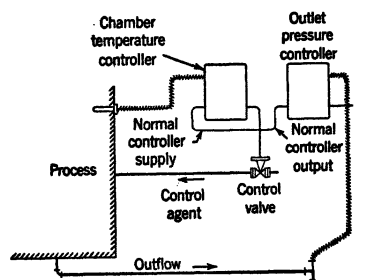
FIG. 10–11. Series-Proportional Control System for Heating Chamber Temperature and Pressure.

During the reaction period, the pressure rises rapidly to its control point and the output pressure drops to a lower value, say 6 lb per sq in. The pressure controller now controls the valve position since the temperature controller cannot produce a higher output pressure than its supply. The controller delivering the lower pressure automatically assumes command of valve position. The chamber pressure is controlled during the reaction period, and when conditions return to normal the temperature again controls the process. By this arrangement the control function is automatically shifted from one controller to the other as required by conditions in the process.

Series control is useful in the control of furnace temperatures where the control function must be shifted from temperature to fuel rate during certain periods and also in air-conditioning control where the control function must shift from temperature to humidity as conditions of the intake air require.

Series-proportional control is not usually accomplished with electric proportional controllers. Greater versatility in series control can be obtained with pneumatic proportional controllers, since the output of one may readily serve as the supply for the other.

DUAL CONTROL AGENTS

In some processes more than one control agent may be needed. The controller must then be arranged so as to regulate two flows into the process instead of one, and the control system includes a single measured variable and two adjusted variables. For that reason, it may be classed as a multivariable control system.

The temperature of plating tanks is controlled by means of dual control agents. The temperature of the circulating water is controlled by admitting steam when the temperature is low, or cold water when it is high. Figure 10–12 illustrates a system where pneumatic proportional control and diaphragm valves with split ranges are used. The steam valve is closed at 8.5 lb per sq in. pressure from the controller, and fully open at 14.5 lb per sq in. pressure. The cold water valve is closed at 8 lb per sq in. air pressure and fully open at 2 lb per sq in. air pressure.

Fig. 10–12. Dual-Agent Control System for Adjusting Heating and Cooling of Bath.

If more accurate valve settings are required, pneumatic valve positioners will accomplish the same function. The zero, action, and range adjustments of valve positioners are set so that both the steam and cold water valves are closed at 8 lb per sq in. controller output pressure. The advantages gained with valve positioners are that standard diaphragm valves are acceptable, settings can be much more accurate, and the action of each valve can be readily adjusted.

For applications having dual agents, duplex controllers are arranged with two complete control systems operating from a single measured variable. Two standard control valves are used, and settings are made by means of the control point of each system. The system may be either electric or pneumatic.

Fig. 10–13. Dual Agent Control System with Two Heating Agents.

Another example of dual control agents, illustrated by Fig. 10–13, is the control of temperature in paper-drying machines. Ordinarily, exhaust steam is utilized for heating, and a valve in the exhaust steam line is set from a temperature controller. Because of varying loads on the steam system, exhaust steam is likely to change considerably in pressure and quality. Then the heat supply of the exhaust steam is supplemented by live steam. Check valves prevent reverse flow,

The connection of the valves is illustrated in Fig. 10–13. The controller pressure is connected to the diaphragms of both valves. The exhaust-steam valve is set to open at 7.5 lb per sq in. and is fully open at 13½ lb per sq in. Both valves are open between 7.5 and 8.5 lb per sq in. controller pressure.

Changes in quality of exhaust steam are automatically corrected by the reset response of a proportional-reset controller. Suppose, for example, that the heat content of the exhaust steam becomes gradually lower. The temperature in the paper machine drops slightly below the control point, and reset response gradually opens the live-steam valve so as to maintain the required heat supply.
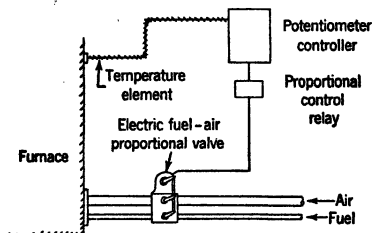
FIG. 10–14. Dual Agent Control System Employing Electric Proportional Control for Adjusting Both Fuel and Air.

The size of each valve should be based upon the relative heating and cooling effect of each agent. The addition or subtraction of heat from the process should be the same with corresponding openings of each valve. The characteristic of each valve should be the same. Linear valve characteristics are preferable since the overall response of the two control agents can more easily be matched.

Dual control agents, and sometimes more than two, are often needed in fuel-fired furnaces burning combinations of oil and air or gas and air. Here again the mechanisms and systems are many and highly individual, but most of them fall into three general classes: those with a single fuel regulated by single valve and with inspirated air; those having a dual or combination valve which adjusts the flow of both fuel and air; and those with metered-ratio control of multiple fuels.

In many simple applications where an exact fuel-air ratio is not required a single fuel is adjusted by a control valve and the air is inspirated with correspondingly greater flow as the fuel flow increases with positioning of the control valve. Also common is a single power unit operating two valves, one in the fuel line and one in the air line. Such an application is illustrated in Fig. 10–14, where electric proportional control is employed for furnace temperature. An electric-motor operator positions both the fuel and air valves, thus maintaining a ratio of fuel to air by virtue of the flow characteristics of each valve. Many of these combination valves are provided not only with a means for adjusting the maximum flow, or valve size, but also with a means for adjusting the flow characteristic to match fuel-air ratio at various settings.

Metered-ratio control is suitable either where exact ratio is necessary, or where multiple fuels are required, or both. For example, many high-capacity furnaces for metallurgical work take three fuels such as oil, gas, and air.[6] The combination of two systems already described provides an interesting problem in the application of automatic control. Metered control will maintain the exact flow of fuel required by the master temperature controller. In addition, exact ratios between fuel flow or flows and air flow must be maintained. Ratio control is thus combined with metered control so that fuel and air are held not only to specified flow rates but also to the desired flow ratio.

When the process can be controlled with a two-position or proportional mode the problem of dual control agents is not difficult. If proportional-reset control is required because of process load changes, proper valve arrangement becomes important to the operation of the control system; smooth changes in flow must be provided by proper selection of valve sizes and proper adjustment of valve operation.

Engineered Control Systems

Many industrial processes require control systems arranged to correlate widely diversified variables affecting the process. In addition, purely functional operations may be incorporated into the control system, not necessarily for control of selected variables but to make the process more continuous in action. In this way the operation of the process is made almost entirely automatic. These systems range, for example, all the way from simple pressure control of curing molds in rubber and plastics manufacture to automatic control of central power stations. Power plants are an excellent example of such coordinated and engineered systems. They are sometimes completely controlled to the extent that operators are required only to maintain automatic systems and handle emergency conditions.

Fig. 10–15. Correlated Control System for Automatic Operation of Reaction Chamber.

Correlated control systems are necessarily different for every application. They must be "tailored" to perform a particular task. A hypothetical case will illustrate what can be done to provide completely automatic operation. The complete unit of Fig. 10–15 is arranged to carry through automatically a complete batch operation.

The purpose of this correlated system is to heat the materials in the

reaction chamber through a controlled cycle and, in addition, control the pressure and cold-water injection during predetermined intervals. This program is accomplished by a process timing mechanism and the necessary temperature, pressure, and auxiliary controllers as follows:

1. Operator manually starts timer to begin program.
2. Timer turns on air supply to time-temperature controller, thereby opening the steam valve to the reaction chamber.
3. Temperature controller raises the temperature of the reaction chamber to 180° F at a linear rate of rise during 2 hours.
4. Temperature of the reaction chamber is held to 180° F for 1 hour.
5. One-half hour after the temperature has reached 180° F, the air supply to the pressure controller is turned on by the timer, thereby opening the air valve.
6. For the remainder of the program the pressure is controlled to 20 lb per sq in. gage.
7. At the end of 1 hour at 180° F, the reaction chamber temperature is lowered to 100° F in ½ hour at a linear rate.
8. When the temperature of the reaction chamber has reached 100° F, the cold-water valve is automatically opened by the timer.
9. Timer shuts off the temperature and pressure controllers.

This example represents the action of a typical engineered system where all control functions are coordinated in order to achieve completely automatic operation. The example is by no means as complicated as some actual applications.

The important coordinating function of the control system is provided by the process timing mechanism. These devices contain a series of cams for operating a set of pneumatic or electric switches. The cams are generally driven by an electric motor operated either on a purely time basis or from external sources.

A process timing mechanism is illustrated in the operation of vulcanizers for rubber tire manufacture. The cycle is as follows:

1. Open steam valve.
2. Raise temperature to vulcanizing point.
3. Hold temperature at vulcanizing point for required period.
4. Close steam valve.
5. Open blow-off valve.
6. Close blow-off valve after required period.
7. Open water cooling valve.
8. Open drain valve after required period.
9. Close water cooling valve after required period.
10. Close drain valve.

It may be noted that only during steps 2 and 3 is there any automatic

control of an independent variable (temperature). All other steps are merely time functions required to provide a complete processing operation.

Many correlated control systems are built around the time-variable control system with an auxiliary electrical timer for performing the required operation at the beginning and end of the cycle. Self-contained units of this nature are used in rubber, food, glass, and tobacco processing to eliminate manual operations which would otherwise be necessary.

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CHAPTER 11

MAINTENANCE OF EXACT CONTROL

For satisfactory, continuous operation, any controller or control system should receive periodic inspection and careful, intelligent maintenance. If, after installation, the quality of control is not wholly satisfactory, a methodical check should be made of all factors which might contribute to faulty operation. Once the difficulty is corrected, care should be taken to prevent its recurrence by positive, direct measures. Dirt, corrosion, wear, and changing conditions are the enemies of exact control.

Controllers are properly classified as precision industrial apparatus, and they should be accorded treatment commensurate with their important role in modern industrial processing plants. However, it is not always possible to locate industrial controllers in clean, corrosion-free atmospheres. On the contrary, many installations are necessarily made under conditions far from ideal.

Although most industrial controllers are designed to withstand severe usage, it is axiomatic that high sensitivity and responsiveness cannot be combined with heavy-duty or "armorlike" construction, particularly in self-operated controllers. Actually, a controller is carefully built to obtain maximum responsiveness with the greatest ruggedness. It is therefore encumbent on the user to provide control equipment with maximum protection from dirt and to insure that operation is not impaired by careless maintenance.

Other factors must be taken into consideration in addition to controller maintenance. Most controllers are power-operated and require an auxiliary power medium such as electrical voltage, air pressure, or hydraulic pressure. The source of auxiliary power must be constant and infallible in operation. Where a large number of controllers are involved, special precautions should be taken to provide a standby supply source.

The process, as well as the control system, requires adequate maintenance. As every experienced control engineer knows, many causes of unsatisfactory performance can be traced to fault in the process. Many operators are prone to overlook this possibility and spend much time investigating the control system rather than the process itself. A partially blocked line, a fouled heat exchanger, or a clogged burner

causes difficulty as readily as a dirty air supply or improperly wired electrical circuit.

Although it is impossible to cover all the details of proper control-system maintenance, a few pertinent points which contribute to efficient operation of controllers must be emphasized.

Temperature Measurement

The quality of control can invariably be improved by increasing the sensitivity and responsiveness of the measuring means. Since the primary element is the first to receive and produce an indication of any changes in the measured variable, the indication must be prompt and of the greatest possible magnitude.

A temperature primary element receives heat by transfer from the surroundings. Any means which aids in increasing this rate of heat transfer decreases the measuring lag and improves control. Although materials of high thermal conductivity speed the transfer of heat, they are of value only if the thermal mass is kept small and losses away from the element are reduced to a minimum.

A copper protecting well has a higher thermal conductivity than an iron or steel well, but, because thermal mass and losses are greater, speed of response is slower. In many applications a ceramic protecting well actually provides greater speed of response than either copper or iron.

With thermocouples, a grounding of the hot junction to the inside of the protecting well provides better conduction. Pencil-type thermocouples in which the iron of the iron-constantan junction forms the protecting well have a high rate of heat transfer although their life is somewhat limited in a corrosive atmosphere.

With pressure thermometer controllers, the speed of response can be greatly improved by reducing the air space between the bulb and well to a minimum, thereby increasing the rate of heat transfer. Positive metal-to-metal contact between well and bulb may be provided by means of thin metal inserts. Filling the space with a suitable fluid or other material has also proved beneficial.

In some applications the speed of response of pressure thermometer controllers can be considerably improved by means of an averaging-type bulb. These bulbs are long in comparison to diameter, sometimes several feet in length. In this way the area is materially increased without a great increase in mass. An averaging bulb may be employed only under non-corrosive conditions. A protecting well increases the measuring lag to a point where there would be little net gain in the averaging-type bulb.

Although protecting wells for both thermometer bulbs and thermocouples are necessary in many applications, the resultant increase in measuring lag must be recognized. Many protecting wells are designed from the standpoint of measuring, not controlling service, since life rather than responsiveness is incorrectly emphasized. The natural inclination to choose a heavy well must be curbed, and economy in replacement costs must be balanced against speed of response.

In dirty or sooty atmospheres and in heavy viscous liquids, deposits may build up on or around a protecting well, decreasing the heat transfer and slowing the response. The outside of the well must be cleaned occasionally; a high velocity of flow past the well aids in scouring the surface.

Pressure Measurement

Flow and pressure measurement have much in common since flow control is usually accomplished through measurement of differential pressure. In addition, fluid inertia and viscous flow in the connecting lines affect the measurement of both flow and pressure equally.

For maximum responsiveness a flow controller requires careful installation. It should never be installed farther from the orifice than is absolutely necessary unless transmission of measurement is used. One of the causes of difficulty in flow control is the inertia of the moving fluids in the connecting lines and meter body. Inertia produces a response similar to transfer lag and results in a wider proportional band. To overcome inertia, greater damping is required in order to obtain stable response, and the measuring lag is thereby increased. Therefore, every available means must be taken to minimize inertia by providing short connecting lines and a small mass of fluid to be moved with changes in flow.

Heavy, viscous fluids, which provide greater damping by virtue of fluid friction to flow, may increase the measuring lag to such a point that considerably overdamped action is obtained. Dynamic error may occur to a serious proportion in flow measurement, and measuring lag must be kept at an absolute minimum if accurate flow control is desired.

The orifice or other flow element is subject to wear or abrasive action which gradually shifts the calibration and therewith the control point. Build-up of deposits on the orifice produces the same result. Periodic inspection and cleaning will eliminate shift of the control point due to these changes in orifice characteristics. Over a period of time dirt and sediment often accumulate in a flowmeter body, and thorough cleaning at regular periods is required.

The measurement of pressure, vacuum, or sometimes of liquid level

is affected by connecting tubing of improper size or length. Small-diameter lines or lines filled with viscous fluids are sure to decrease the speed of response. Clogging of the connecting line also increases the measuring lag.

Measurement transmission systems should always be considered when the controller must be located at a distance from the primary element. They not only make installation of the complete system easier but also provide faster and more accurate measurement.

With self-operated measuring means, as in flow or pressure thermometer controllers, the friction of the pen against the chart in a recording controller must not be overlooked in checking the performance of the control system. Pen friction or any other friction in the measuring means is decidedly detrimental to control since it allows the measured variable to change without detection by the control system. A recording pen should be adjusted to draw the finest line producing a satisfactory control record.

Electric Controllers

Although electric and electronic controllers are generally designed to be independent of line voltage and frequency changes, it is advisable to draw upon the best power source available. Power failure must be safeguarded since the controller cannot operate without electrical power. An overloaded line or a line on which overloads are likely to occur should be avoided if another source is available.

Fig. 11-1. Suggested Method of Installing Electric Controller.

The arrangement shown in Fig. 11-1 is common. A line switch should be installed so that power may be cut off the control system. Sometimes all switches are omitted from the controller, only a line switch being relied upon to close down the control system. Fuses of proper size should protect the control equipment. In some applications a transformer is required when the control equipment must be operated from a higher voltage line. An isolation transformer may be needed with electronic controllers if one side of the line is to be grounded. A constant-voltage transformer may be necessary if adjacent heavy machinery is likely to cause excessive voltage surges. All wiring should

be installed in the proper type of conduit, which must be waterproof or splashproof in wet locations.

Two- or three-position controllers of the electric type generally require little maintenance for satisfactory operation. Open metallic contacts or relays, however, sometimes need to be inspected and cleaned in an electric circuit of any type. It is likely that the motor-operated valve or other element will require more attention than the controller.

Electric proportional controllers require careful maintenance for their best performance. Slide wires or other mechanisms for signaling position and their wiping contacts must be kept clean and have proper contact pressure. On temperature applications the controller lag is determined by the sensitivity of the system, and in order to reduce the lag to a minimum sensitivity must be maintained as high as possible.

Electronic control equipment always utilizes electronic tubes or other electrical components of the type which may age with use. It is not desirable to wait for the tubes to fail before replacing them. Tubes for such equipment are generally of standard types, and a regular replacement schedule may save the cost of a shutdown.

Many electronic control systems which include amplification are made with certain adjustable circuit components. These sometimes require readjustment, and instructions as to their use should be carefully followed in order to maintain satisfactory operation.

In servicing electronic equipment it is important that the operation of the circuit be understood. Without this knowledge it is better to obtain experienced aid in locating trouble. As a caution, electronic circuits often have voltages much higher than line voltage, and care must be exercised in handling such equipment.

Electric controllers generally require very little lubrication. When oil is required it is important that neither too little nor too much be applied. Oiling should be done in strict accordance with the manufacturer's instructions.

Pneumatic Controllers

Pneumatic controllers require very little maintenance. The various pivots and links in the control system should be free of friction and in good working order. Oil should never be applied except by specific instruction of the manufacturer.

Connections in pneumatic or hydraulic control systems must be absolutely tight. Loose connections not only waste air or oil but also produce unsatisfactory operation. When installing a pneumatic system the connections should always be checked for leakage.

The size of connecting air lines leading from the controller to a final

control element is an average between large diameter, giving large volume, and small diameter, causing large resistance to flow. In order to maintain maximum speed of response, the optimum internal diameter for connecting lines is approximately ¼ inch.

Volume at the output connection of pneumatic controllers should be kept as small as possible to insure a minimum of controller lag. The volume of the diaphragm valve top greatly influences the controller lag and should not be greater than is necessary for valve operation. If the volume of the valve top is large, a booster relay, or a valve positioner will aid in reducing controller lag.

On the other hand, if the volume of the output connection is small, the controller will be underdamped and may oscillate slightly. Tubing is much more effective in eliminating oscillation than a volume tank, and, moreover, it does not appreciably increase controller lag. Generally, 30 or 40 feet of tubing is sufficient to damp out oscillation without affecting speed of response.

In order to eliminate the presence of foreign material inside the lines, particularly the air-supply line, non-ferrous materials like copper tubing or brass pipe are preferred. Iron or steel pipe will ultimately rust, and small solid particles of material may cause difficulty.

Small passages in pneumatic controllers are usually protected by small waste or screen filters to keep out any foreign material. These filters require periodic inspection; otherwise pressure drop may become excessive and the pneumatic system may fail to operate.

A pressure regulator is generally placed at each controller to provide a constant supply pressure. Most pneumatic controllers are designed so that supply-pressure changes of moderate magnitude do not shift the control point to any appreciable extent. However, the simpler types of pneumatic controllers without feedback may be subject to changes in supply pressure.

Air Supply for Pneumatic Systems

The greatest problem connected with pneumatic controllers is the maintenance of a clean, dry air supply at constant pressure. Moisture, oil, corrosive liquids, or foreign particles carried into the pneumatic system from the air supply will eventually cause trouble. Pneumatic controllers operating with clean air require virtually no cleaning or maintenance.

An air-supply system similar to that shown in Fig. 11–2 may be appropriate where a large amount of control equipment is involved, as in an oil refinery or chemical plant. An intake filter is located outside the compressor room through which the air passes to the com-

pressor. A steam-driven compressor of desired capacity at a delivery pressure between 150 and 200 lb per sq in. gage is common. The air passes into a storage tank, through a pressure regulator holding the pressure to 100 lb per sq in., and then to an aftercooler. The after-
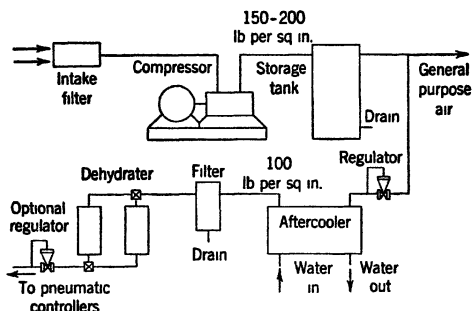
Fig. 11–2. Complete Air-Supply System for Large Installation of Pneumatic Control Equipment.

cooler or condenser requires a water supply. A porous stone filter removes oil, and a dehydrator completes the removal of moisture. A second pressure regulator is sometimes added to provide constant reduced pressure in the supply lines leading directly to the control rooms for a group of pneumatic control equipment.
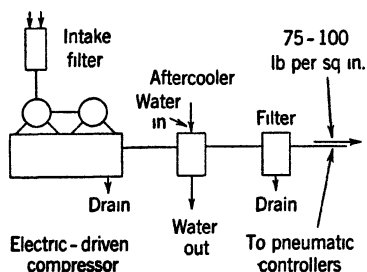
Fig. 11–3. Air-Supply System for Small Installation of Pneumatic Control Equipment.

In the majority of small installations an arrangement as shown in Fig. 11–3 is satisfactory, especially where excessively low or high temperatures are not encountered and where the intake air is relatively dry. An intake filter removes dust and solids from the intake air. A compressor with attached storage tank contains a pressure-actuated switch to maintain relatively constant pressure in the tank. An after-cooler and large-capacity filter removes excess oil, moisture, and solids.

Solids are removed most easily by a suitable intake filter at the compressor. The intake filter should be located at a point where it may gather the cleanest air possible; this point is generally outside the

building housing the compressor. Edge type, gauze, or waste filters may be placed at various points in the system to separate solid particles or foreign matter. Clean connecting lines of brass or copper are desirable to eliminate rust.

Moisture is usually adequately removed with a storage tank of proper size and a compressor aftercooler. Where moisture conditions are severe, chemical or electrical driers in addition to aftercooling may be required to remove moisture more completely. Chemical driers containing silica gel, activated alumina, calcium chloride, alcohol, glycol, or other drying agents are common equipment. When the drying agent requires replacement or regeneration, a two-section dehydrator should be installed, only one section being used at a time while the other section is being regenerated. If the drying agent is a liquid, an additional unit may be needed for removing the drying agent carried over in the dry air. Lines should be trapped and installed on a slope to drain properly. Check valves should be inserted if there is a possibility that liquids or other gases might back into the lines from other sources.

Oil should be prevented from entering into the system. The air compressor should never be overloaded, since it will pump more oil when running at high loads. A water-sealed rotary compressor where no oil comes in contact with the air is sometimes preferred. The air compressor should receive careful maintenance in order to obtain high efficiency at all times. An aftercooler aids in removing oil vapor as well as moisture. Filters of the porous-stone, fuller's earth, centrifugal, or waste type effectively remove larger suspensions of oil. Solvent-type filters may be used to remove oil, although it may then be necessary to remove the solvent dissolved in the air.

An individual filter should be placed at each controller to remove any remaining moisture, oil, or solids just before the air enters the controller. The waste-type filter is most common and is quite satisfactory under ordinary conditions. An individual small-capacity pressure regulator should accompany each controller or group of controllers in order to provide proper pressure regulation.

Moisture-free compressed air does not freeze even at subzero temperatures. Therefore, methods of effectively removing moisture also aid in avoiding trouble due to freezing of air lines. Anti-freeze injection systems are not adequate since the total amount of entrained fluid in the air may be increased.

Locating the air lines so as to avoid low-temperature areas as much as possible is of considerable aid in eliminating either condensation or freezing. It may be possible to maintain the temperature of a

group of air lines above freezing by means of a steam tracer. Burying the lines in the ground below the frost line, with suitable traps to remove condensate, may be effective.

It is sometimes possible to substitute other compressed gases than air for the operation of pneumatic controllers where an adequate air supply cannot be provided. Such gases must not bring about any corrosive action or deterioration of the parts of the control system. For strictly emergency short-time operation of a single control system bottled nitrogen may be feasible.

Final Control Element

The power unit and final control element require careful installation and checking in order to maintain desirable operating characteristics. It is sometimes necessary on severe applications to remove and service a control valve at each shutdown of the process.

A diaphragm control valve should be installed in a vertical position whenever possible. An electric motor valve should be installed with the motor shaft horizontal. When the power unit is furnished separately from the valve, it should be mounted rigidly so that there is no relative motion between the power unit and valve. Such motion is a common source of trouble and causes insensitiveness in the positioning of the valve. The power unit should also be connected rigidly to a damper, louver, or any other final element. Lost motion at the connection must be avoided at all costs.

When connecting a power unit to a damper or louver, advantage should be taken of any characterization that can be obtained by means of linkage arrangement. It is often quite simple to plan the linkage to provide small motion of the valve at the closed position and large motion at the open position. Larger motion at the closed position than at the open position should never be permitted.

It is equally important to obtain smooth motion of the final element. If either the power unit or the valve causes rough or uneven calibration of flow against applied pressure or voltage at the average operating point, the cause should be corrected or exact control will be difficult to maintain.

Nearly all control elements have some kind of packing gland. Friction exists at this point and must be overcome by the power unit when motion is required. A lubricator is applied on control valves in order to maintain a seal at the packing. Careful adjustment of the packing gland is required so that neither leakage nor sticking occurs. Sticking and leakage should always be eliminated by lubrication, not by tighten-

ing the packing gland. Excessive lubrication, however, may solidify the packing.

On the average sliding-stem control valve the force of friction is between 3 and 12 lb. On high-temperature or -pressure applications, and where corrosive fluids are handled, the force of friction may be as high as 25 lb or even more. If the power unit does not have a high applied force, an excessive dead zone will be caused. A positioner is always required on pneumatic diaphragm valves when friction is excessive.

Electric motor valves generally need lubrication at selected points. The recommended lubricant should be applied regularly. Limit switches and slide wires of electric motor valves should be kept clean. An inoperative limit switch may cause damage.

An undersize final element does not permit the controller to correct for excessive deviation; an oversize one may have a large dead zone. Excessive undersizing and oversizing should be corrected. If a valve operates near the closed position, wire drawing may occur, causing excessive wear on the seats. Not only will a valve with worn seats have a high leakage but also the characteristic may be considerably altered. It may be necessary to rebuild the control valve periodically to maintain its operating condition.

The pressure differential characteristic of a valve, damper, or louver should be carefully checked if undersizing seems apparent. It is possible that line losses may be so large that the required pressure differential is not present.

A pneumatic diaphragm valve or any other pneumatically operated unit should never be operated below 1 or preferably 2 lb per sq in. pressure. In other words, initial suppression should be used so that the valve is either fully closed or open at 2 lb per sq in. and begins to open or close on a further increase of pressure. A 2 to 14 lb per sq in. operating range is generally preferable.

Process Arrangement

Difficulties in obtaining satisfactory control may be due to process characteristics as well as to improper operation of the control system. Any analysis of the process should include an inspection of the controlled variable, a tabulation of the various rates and magnitudes of load changes, a study of the degree of process lags, and a careful observation of the relation among these factors.

Many times it is difficult to assign a relationship to the controlled variable and the desired state of balance of the process. Even though

the controlled variable may be satisfactorily controlled, the state of balance of the process may be anything but satisfactory. For example, take the hypothetical case of controlling the reaction of chemicals in a pressure vessel. Suppose that we measure and control the pressure. If the pressure is steady, the temperature is constant (assuming no exothermic or endothermic reactions). When the temperature is constant, the rate of reaction is constant. The quality or composition of the processed materials should therefore be correct. However, if any one of these four steps should break down or become non-proportional, then the controlled variable loses its meaning with respect to control of the process.

For this reason it is important to control directly from final product or a final balance quantity of the process whenever possible, in order to eliminate the possibility of any variance between the controlled variable and the control conditions of the process.

Maintaining the balance of the process is difficult when the load or any of the uncontrolled variables associated with the process are subject to frequent and fast changes. The deviation of the controlled variable is very nearly in direct proportion to the rate of the changes. It is often necessary to add a controller to the changing variable in order to eliminate its effect on the control of the main process variable. For example, in large kilns and furnaces it is nearly always necessary to control the pressure in the furnace in order that furnace temperature may be sufficiently stabilized.

The effect of supply variations and disturbances should be eliminated. Metered control is the best means of eliminating disturbances due to variations in the pressure of the fuel supply. When the Btu value of the fuel varies, as for example when mixed natural and propane gases are used, it is sometimes necessary to provide a mixing chamber so that changes in Btu value are made gradually.

It is often possible to improve the control of the process by making simple alterations or by redesigning to obtain greater demand capacity, smaller transfer lag, or shorter dead time. Unless these steps are undertaken with very careful consideration of the effects of dynamic process reactions, the improvement may be handicapped by other disadvantages. One would not increase the demand capacity, for example, if a large transfer lag was thereby introduced.

Examples of improvements by rearranging the process are often found in heat exchangers. The relocation of the supply line or the moving of a baffle may bring marked improvement in the control of demand temperature. For instance, in the control of plating tanks serious transfer lag is eliminated by controlling the temperature of the supply

medium and adding forced circulation of the supply medium around the tank.

Dead time must be kept to a minimum in the controlled system. This type of lag should always be investigated to determine whether it can be reduced or eliminated. As we have seen, dead time is the most serious lag in automatic control.

Transfer lag is caused by a second capacity in the process or by distributed capacity in heat transfer. When reducing transfer lag, the second largest or third largest capacity should be eliminated. The largest capacity, no matter where it is located in the process, should remain to provide stability. Large capacity of itself is not detrimental to automatic control provided that all associated capacities are negligible by comparison. Transfer lag caused by large temperature or pressure differences between capacities can sometimes be reduced by a rearrangement of the supply medium.

Maintenance and Servicing Methods

Responsibility for maintenance of automatic control systems should be centralized at one point in the plant organization, whether the amount of work is small or large. This centralization makes regular maintenance possible, provides for prompt emergency service, and is also a means for coordinating all the experience gained from installing and operating control systems.

An instrument department, whether it is composed of only a few or a large number of instrument men, is generally made responsible for the operation of instruments and controllers. In order to meet the requirements for adequate handling of control systems, an instrument department must be acquainted with the theory of measurement, the theory of control, the design of instrument and control equipment, and the operation of process or plant. An understanding of these fundamentals is essential in operating the types of control systems found in modern industrial plants. Industrial technology is rapidly extending measurement and automatic control in order to increase efficiency and output, and, as control systems become more complex, knowledge of basic principles becomes more important.

An instrument department, large or small, is generally divided into at least the following sections: a pyrometer or electrical section, a flowmeter or pressure section, and a library. In larger groups a thermocouple section, a small-parts section, and a heavy-machinery section may be required. The instrument department should be given adequate housing, tools, and test equipment.

Individual requirements dictate the organization of the department,

but the library and files section should not be overlooked. A library should include all data and information necessary for use or reference in the instrument department. Records of equipment, instruction manuals, manufacturers' data, and repair parts lists should be carefully maintained.

Full records should be kept on all control equipment. Often it is necessary to know the date of an installation, the results of a calibration check, or whether any renewal of parts has been made. Instrumentation difficulties can often be traced to the omission of cleaning of a filter, inspection of a bearing, or some similar detail. Usually all information concerning a particular piece of control equipment is entered on a form for the purpose of providing a complete history.

Regular schedules should be maintained because most controllers and their associated equipment require some kind of monthly or yearly attention to details of oiling or cleaning. In small industrial plants the periodic maintenance required for a few controllers is not time-consuming but it is easily overlooked. In large industrial plants where many hundreds of controllers may be serviced, regular check-off schedules insure periodic attention.

When the number of controllers is sufficient, it is an accepted arrangement to divide all types of maintenance work equally among instrument personnel. Often a specific area or process unit is assigned to each instrument engineer but the entire personnel is rotated among the various areas so as to increase familiarity with all types of problems. An electronic or valve " expert " is too likely to be unavailable when urgently needed. The knowledge required for all the various types of control systems is not so greatly varied but that the average good mechanic can be trained to cope with the problems.

It is virtually certain that electronic means of operation in both measuring and control means will become increasingly important. Pneumatic and electric means of operation will be supplemented by electronic systems. Consequently, the experience and training required for handling pneumatic, hydraulic, and electric circuits should be supplemented by training in electronic methods.

Familiarity and experience with instrument and controller operation are best gained through actual use. Valuable background and information on theory and practice can be obtained by utilizing the training divisions established by many instrument and controller manufacturers. Large industrial users also have established training methods. Many engineering societies, like the American Society of Mechanical Engineers and the American Institute of Electrical Engineers, devote a division or section to the science of measurement and control, and

much valuable information may be obtained through their activities. Local groups also may contribute to the training of instrument engineers. The combination of training and experience is an asset in the maintenance of automatic control systems.

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GLOSSARY OF TERMS USED IN AUTOMATIC CONTROL

The following terms and their definitions have come into use in the science of automatic control. These definitions are offered subject to revision by further issuance and adoption of terminology by the A.S.M.E. Industrial Instruments and Regulators Division Committee on Terminology.

AUTOMATIC CONTROL. Automatic maintenance of balanced conditions within a process.

AUTOMATIC CONTROLLER. Mechanism which measures the value of a variable quantity or condition and operates to correct or limit deviation of the measured value from a selected reference.

AUTOMATIC CONTROL SYSTEM. An automatic controller including primary element and final element in which the action is operated on or influenced by other control mechanisms or automatic controllers external to the first automatic controller.

AUTOMATIC REGULATOR. *See* Automatic controller.

AUTOMATIC RESET. *See* Reset response.

CAPACITY. Change in stored energy or material required to produce unit change in potential or level.

CASCADE CONTROL SYSTEM. *See* Metered control system.

CONSTANT-SPEED FLOATING MODE. *See* Single-speed floating mode.

CONTROL. *See* Automatic control.

CONTROL AGENT. Process energy or material of which the manipulated variable is a condition or characteristic.

CONTROL CIRCUIT. *See* Controlled system.

CONTROL POINT. Selected reference value of controlled variable which it is desired to maintain.

CONTROL SYSTEM. *See* Automatic control system.

CONTROLLED MEDIUM. Process energy or material in which a variable is controlled.

CONTROLLED SYSTEM. Interconnected process, controller, or controllers, and control mechanisms wherein a single designated variable is maintained within limits.

CONTROLLED VARIABLE. Quantity or condition which is measured and controlled.

CONTROLLER ADJUSTMENT. Manually adjustable characteristic of an automatic controller for varying relationship between controlled variable and controller response.

CONTROLLER LAG. Retardation or delay in response of final control element to change in controlled variable at the controller.

CONTROLLER RESPONSE. Output signal or impulse from an automatic controller.

CONTROLLING MEANS. Elements of an automatic controller which are involved in producing a corrective action.

CORRECTIVE ACTION. Controller action initiated by deviation and resulting in variation of the manipulated variable.

CORRELATED CONTROL SYSTEM. An automatic controller including primary element and final element in which one or more measured variables are controlled in relation to another measured variable or in relation to time.

CYCLING. Periodic change of controlled variable from one value to another.

DEAD TIME. Any definite delay period in the measuring means, controlling means, or process.

DEAD ZONE. Largest range of values of controlled variable to which the measuring means or controlling means does not respond.

DEMAND CHANGE. *See* Load change.

DEPARTURE. *See* Deviation.

DERIVATIVE MODE. Controller action in which there is a continuous linear relation between derivative function of controlled variable and position of final control element. (Linear controller scale is assumed.)

DEVIATION. Difference at any instant between value of controlled variable and control point.

DIFFERENTIAL GAP. Two-position controller adjustment: smallest range of values through which controlled variable must pass in order to move final control element to both its fixed positions. (Linear controller scale is assumed. Expressed in percentage of controller scale.)

DISTANCE-VELOCITY LAG. *See* Dead time.

DRIFT. Wandering of controlled variable in which its value aimlessly departs from control point.

DROOP. *See* Offset.

DROOP CORRECTION. *See* Reset response.

DYNAMIC ERROR. Difference between true value of a quantity or condition changing with time and the value indicated by a measuring means.

ERROR. *See* Deviation.

FINAL CONTROL ELEMENT. Portion of controlling means which directly determines the value of manipulated variable.

FLOATING MODE. Controller action in which there is a predetermined relation between value of controlled variable and rate of motion of final control element, the direction of motion corresponding to the direction of deviation.

FLOATING RATE. Proportional-speed floating controller adjustment: rate of motion of final control element corresponding to a specified deviation. (Linear controller scale is assumed. Expressed in percentage motion per minute per per cent deviation.)

FLOATING SPEED. Single or multispeed floating controller adjustment: rate of motion of final control element. (Expressed in percentage motion per minute.)

HUNTING. *See* Cycling.

INACTIVE NEUTRAL. *See* Differential gap.

INSTRUMENT. Mechanism for automatically measuring and indicating the value of a quantity or condition.

INTEGRAL RESPONSE. *See* Reset response.

INVENTORY. *See* Capacity.

LAG. Retardation or delay of one physical condition with respect to some other condition to which it is closely related.

LEAD COMPONENT. *See* Rate response.

LIMIT CONTROL SYSTEM. *See* Series control system.

LOAD CHANGE. Change in process conditions which requires a change in the average value of manipulated variable to maintain the controlled variable at the desired value.

LOAD ERROR. *See* Droop.

MANIPULATED VARIABLE. Quantity or condition which is varied by the automatic controller so as to affect the value of controlled variable.

MASTER CONTROLLER. In a metered control system, that automatic controller which adjusts the control point of another automatic controller.

MEASURED VARIABLE. Quantity or condition the value of which is automatically ascertained by an instrument or an automatic controller.

MEASUREMENT. Act of ascertaining the value of a quantity or condition.

MEASURING LAG. Retardation or delay in response of measuring means of an instrument or an automatic controller to changes in measured variable.

MEASURING MEANS. Elements of an instrument or automatic controller which are involved in ascertaining value of measured variable.

MEASURING SYSTEM. *See* Measuring means.

METER. *See* Instrument.

METERED CONTROL SYSTEM. An automatic control system in which the automatic controller operates a second automatic controller for adjusting the value of the manipulated variable.

MODE OF CONTROL. Systematic method of action of a controller.

MODULATING MODE. *See* Proportional mode.

MULTIAGENT CONTROL SYSTEM. An automatic control system in which two or more manipulated variables are adjusted by a single automatic controller.

MULTIPOSITION MODE. Controller action in which a final control element is moved to one of three or more predetermined positions, each corresponding to a definite range of values of controlled variable.

MULTISPEED FLOATING MODE. Controller action in which a final control element is moved at two or more rates, each rate of motion corresponding to a definite range of values of controlled variable, and the direction of motion corresponding to the direction of deviation.

MULTIVARIABLE CONTROL SYSTEM. An automatic control system in which there is more than one controlled variable, or more than one manipulated variable.

NEUTRAL ZONE. Floating controller adjustment: predetermined range or values of controlled variable in which no corrective action occurs. (Linear controller scale is assumed. Expressed in percentage of controller scale.)

OFFSET. Sustained deviation obtained with two-position, multiposition, or proportional mode of control.

ON-OFF MODE. *See* Two-position mode.

OPEN-AND-SHUT MODE. *See* Two-position mode.

OSCILLATION. *See* Cycling.

POSITIONING MODE. Controller action in which there is a predetermined relation between value of controlled variable and position of final control element.

POWER-OPERATED CONTROLLER. Automatic controller with either power-operated controlling means or power-operated measuring means.

POWER-OPERATED CONTROLLING MEANS. Type of controlling means in which the energy transmitted through the primary element is either supplemented or amplified for operating the final control element by employing energy from another source.

POWER-OPERATED MEASURING MEANS. Type of measuring means in which the energy transmitted through the primary element is either supplemented or amplified for operating the indicating mechanism of an instrument or the controlling means of an automatic controller by employing energy from another source.

POWER UNIT. Portion of controlling means which applies power for operating final control element.

PRIMARY MEASURING ELEMENT. Portion of measuring means which first utilizes energy from controlled medium to produce a condition representing the value of controlled variable. Condition produced by primary element is usually pressure, force, or motion.

PROCESS. Operation or series of operations in which the value of a quantity or condition is controlled. It includes all functions which directly or indirectly affect value of controlled variable.

PROCESS EQUIPMENT. Physical apparatus, exclusive of automatic control equipment, for carrying out a process.

PROCESS LAG. Retardation or delay in response of controlled variable at point of measurement to a change in value of manipulated variable.

PROCESS LOAD. Sum, taken at any instant, of the energy or material requirements of the process resulting in a specific value of manipulated variable.

PROCESS LOAD CHANGE. *See* Load change.

PROCESS MEDIUM. Any energy or material which is supplied to or taken from a process and which directly or indirectly affects the value of the controlled variable.

PROCESS REACTION RATE. Maximum rate of change of controlled variable caused by a specified, sudden change in value of manipulated variable.

Process Self-Regulation. Sustained reaction inherent in the process which assists or opposes the establishment of equilibrium.

Process Variable. *See* Variable.

Program Control System. *See* Time-variable control system.

Proportional Band. Proportional controller adjustment: range of values of controlled variable which corresponds to full operating range of final control element. (Linear controller scale is assumed. Expressed in percentage of controller scale.)

Proportional Mode. Controller action in which there is a continuous linear relation between value of controlled variable and position of final control element. (Linear controller scale is assumed.)

Proportional Plus Floating Mode. Controller action in which proportional and floating actions are combined additively.

Proportional-Reset Mode. Controller action in which proportional and proportional-speed floating actions are combined additively in such a manner that the proportional band adjustment affects both actions simultaneously.

Proportional-Reset-Rate Mode. Controller action in which proportional, proportional-speed floating, and rate response actions are combined additively in such a manner that the proportional band adjustment affects all three actions simultaneously.

Proportional-Speed Floating Mode. Controller action in which there is a continuous linear relation between value of controlled variable and rate of motion of final control element, the direction of motion corresponding to the direction of deviation. (Linear controller scale is assumed.)

Rangeability. Ratio of maximum flow to minimum controllable flow through a final control element.

Rate Response. Controller action used in conjunction with the proportional mode in which there is a continuous linear relation between rate of change of controlled variable and position of final control element. The magnitude of rate response is directly related to rate time and inversely related to proportional band. (Linear controller scale is assumed.)

Rate Time. Proportional-reset-rate controller adjustment: advance of proportional response caused by addition of rate response. (Linear controller scale is assumed. Expressed in minutes.)

Ratio Control System. An automatic control system in which value of the variable is controlled in a predetermined relation to value of another measured variable.

Reaction Curve. Change with time of a measured variable resulting from a sudden change in manipulated variable.

Recovery. Change with time of a controlled variable resulting from a sustained or temporary change in process load.

Regulation. *See* Automatic control.

Regulator. *See* Automatic controller.

Reset Rate. Proportional-reset controller adjustment: number of times per minute that proportional response is duplicated by the proportional-speed floating response. (Linear controller scale is assumed. Expressed in number per minute.)

Reset Response. Controller action used in conjunction with proportional mode in which there is a continuous linear relation between value of controlled variable and rate of motion of final control element. The magnitude of reset response is directly related to reset rate and inversely related to proportional band. (Linear controller scale is assumed.)

Resistance. Potential difference required to produce unit change in flow.

Secondary Controller. An automatic controller in which the control point is automatically and continuously adjusted from an external source.

Self-Operated Controller. Automatic controller with self-operated measuring means and self-operated controlling means.

Self-Operated Controlling Means. Type of controlling means in which all the energy necessary to operate the final control element is derived from the measuring means.

Self-Operated Measuring Means. Type of measuring means in which all the energy necessary to operate the indicating mechanism or controlling means is derived from the controlled medium through the primary element.

Sensitivity. *See* Proportional band. Sensitivity is the inverse of proportional band.

Series Control System. An automatic control system in which the value of the manipulated variable is determined by either one of two controlled variables.

Servo-Operated Controller. *See* Power-operated controller.

Set Point. *See* Control point.

Single-Speed Floating Mode. Controller action in which final control element is moved at a single rate, the direction of motion corresponding to the direction of deviation.

Stability. State of controlled variable in which the variable does not cycle, or cycles with decreasing amplitude.

Static Error. Difference between true value of a quantity or condition not changing with time, and the value indicated by a measuring means.

Supply Change. *See* Load change.

Throttling Mode. *See* Proportional mode.

Throttling Range. *See* Proportional band.

Time-Variable Control System. An automatic control system in which value of the controlled variable is controlled in a predetermined relation to time.

Transfer Lag. Retardation, not delay, in response of controlled variable caused by existence of distributed capacity or two or more separated capacities in a controlled system.

TRANSMISSION LAG. Retardation or delay caused in transmitting a measurement of variable from primary element to controller.

TRANSPORTATION LAG. *See* Dead time.

TURNDOWN. Ratio of normal maximum flow to minimum controllable flow through a final control element.

TWO-POSITION DIFFERENTIAL-GAP MODE. Controller action in which a final control element is moved from one of two fixed positions to the other at a predetermined value of controlled variable, and subsequently to the other position only after the variable has crossed a range of values to a second predetermined value.

TWO-POSITION MODE. Controller action in which a final control element is moved from one of two fixed positions to the other at predetermined values of the controlled variable.

UPSET. *See* Load change.

VARIABLE. Quantity or condition associated with a process, the value of which is subject to change with time.

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